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A TEXT-BOOK

OF

MECHANICAL DRAWING

AND

ELEMENTARY

MACHINE DESIGN.

BY

Simpson

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PREFACE.

To properly prepare students for advanced machine design it has been found necessary to introduce a course designed to apply the principles of mechanical drawing to the solution of practical problems in machine construction and to familiarize the student with the arrangement and proportions of the most important machines and their details recognized by competent engineers to be the best practice of the present time.

It is essential to intelligent study and an economical expenditure of time and labor that, before attempting to design a new machine or improve an old one, the student should post himself with all possible information concerning what has *already been done* in the same direction.

To this end the present work has been prepared. In it we have attempted to show what is the best United States practice in the design and construction of various machines and details of machines, using rules and formulæ whenever feasible in working out practical problems.

In addition to this will be found the latest and most approved drafting-room methods in use in this country, without which most drawings would be practically useless. Up to the present time no text-book that we know of has been

published in the United States that could in the best way fill the need as explained above.

Books of a somewhat similar nature have been published in Great Britain, showing that the same need has been felt there as here. These books, modified to suit American practice, have been used to some extent in this country because they were the best to be had, but are not by any means all that can be desired for our purpose in their present form.

While preparing this course for the sophomore students in Sibley College the authors endeavored to secure samples of the *actual machines* or *parts of machines* as collateral in illustrating the exercises given in the book, with a result that in our drafting-rooms we have many examples of modern machine construction placed convenient to the students' hands, so that they may examine and handle the *actual thing* itself while solving the problems in drawing and designing. This we believe of great importance in the study of machine design and construction, because few are able to describe a machine even with the assistance of a drawing so well as to enable the student to conceive it in his mind as it actually is.

The preparation necessary for the proper understanding and execution of the problems contained in this book is as follows: use of instruments, instrumental drawings applied to drawing geometrical problems in pencil and ink, thorough knowledge of the conventional lines, hatch-lining and colors for sections, mechanical and free-hand lettering, orthographic projection in the third angle, isometrical drawing—in brief all that is contained in "A Course in Mechanical Drawing," by John S. Reid, published by John Wiley & Sons, New York.

In the preparation of the drawings for this work we are

indebted to many of the leading engineering firms of this and other States, who have kindly supplied us with drawings and *samples* of the latest and best practice of the day. Our thanks are especially due to the Dodge Manufacturing Company, the Detroit Screw Works, the Buckeye Engine Co., the United States Metallic Packing Co., the National Tube Works, the Ridgeway Dynamo & Engine Co., the Murray Gun Works, Henry R. Worthington, Robt. Pool & Sons, the Baldwin Locomotive Works, the Schenectady Locomotive Works, the American Pulley Co., the Hyatt Roller Bearing Co., the MacIntosh and Seymour Engine Co., and many others.

Our acknowledgments are also due to many of the best authorities on the different subjects treated, among which may be mentioned Thurston's "Materials of Construction," A. W. Smith's "Machine Design," Klein's "Machine Design," Unwin's "Machine Design," Barr's "Boilers and Furnaces," Peabody and Miller's "Steam Boilers," Low and Bevis's "Drawing and Designing," John H. Barr's "Kinematics," Thurston's "Steam Boilers," Reuleaux's "Constructor," the "Proceedings of the American Railway Master Mechanics' Association," etc., etc.

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SUGGESTED COURSES.

FALL TERM.

1. Ex. 1, 3, 4, 5. 6, 7, 10, 12, 13, 15, 19, 22, 24, 26, 29, 30, 32, 34, 38, 40, 46, 51.
2. Ex. 2, 3, 4, 5, 6, 8, 10, 11, 14, 16, 18, 20, 24, 27, 29, 31, 33, 35, 39, 41, 47, 51.
3. Ex. 1, 3, 4, 5, 6, 8, 9, 12, 13, 17, 19, 22, 23, 25, 29, 30, 32, 34, 38, 42, 48, 51.
4. Ex. 2, 3, 4, 5, 6, 7, 9, 11, 14, 15, 18, 21, 24, 28, 29, 31, 33, 36, 38, 43, 49, 51.
5. Ex. 1, 3, 4, 5, 6, 8, 10, 12, 13, 16, 19, 22, 23, 26, 29, 30, 32, 34, 38, 44, 50, 52.
6. Ex. 2, 3, 4, 5, 6, 7, 9, 11, 14, 17, 18, 21, 24, 27, 29, 31, 33, 37, 39, 45, 50, 52.

FALL TERM CONTINUED.

1. Ex. 52, 54, 59, 64, 68, 73, 77, 86, 89, 90, 93.
2. Ex. 52, 55, 60, 65, 70, 74, 84, 87, 90, 92, 94.
3. Ex. 52, 54, 61, 66, 71, 75, 85, 88, 90, 91, 93.
4. Ex. 52, 56, 62, 67, 70, 76, 84, 86, 90, 92, 94.
5. Ex. 53, 57, 63, 68, 71, 77, 85, 87, 90, 91, 93.
6. Ex. 53, 58, 64, 69, 72, 76, 84, 88, 90, 92, 94.

WINTER TERM.

1. Ex. 95, 97, 99, 101, 103, 106, 108, 111, 113, 117, 119, 121, 124, 130, 136, 139, 142, 145, 147, 149.
2. Ex. 96, 98, 100, 102, 104, 105, 107, 112, 114, 118, 120, 122, 125, 131, 137, 140, 143, 146, 148, 149.
3. Ex. 95, 97, 99, 101, 104, 107, 110, 112, 115, 117, 121, 123, 126, 132, 138, 139, 142, 145, 147, 140.
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5. Ex. 95, 97, 99, 101, 104, 105, 108, 111, 113, 116, 120, 121, 128, 134, 137, 140, 142, 145, 147, 149.
6. Ex. 96, 98, 100, 102, 106, 107, 110, 112, 115, 117, 119, 122, 129, 135, 136, 138, 143, 146, 148, 149.

DRAWING AND DESIGNING.

INTRODUCTORY INSTRUCTIONS.

MECHANICAL drawing as applied to machine drawing and design consists of the application of descriptive geometry or orthographic projection to the delineation of machines and parts of machines (modified sometimes by certain conventions) generally recognized by experienced draftsmen.

It is comparatively a simple matter for any person of average intelligence to acquire the ability of making a fairly accurate mechanical drawing of a machine, given the dimensions, but it is altogether a different and more difficult problem to determine those dimensions that will give the best form and proportion to the different parts of the machine as will enable them to properly perform the functions for which they are intended in accordance with the strength of the material of which they may be made.

A mere copy of a drawing unaccompanied by some means for compelling the student to study (1) the form and proportions given and reasons for same or (2) the illustrations of some principle connected with projection is not of much moment in the study of machine drawing and design. But a problem in drawing and design illustrated by a drawing of the object, representing the best modern practice and requiring the calculation of the proportions of the different parts

from rules and formulæ, will induce the student to think, and tend to develop any natural ability he may have in this direction. It has been the aim of the authors in the arrangement of problems to accomplish this purpose in the highest degree possible.

The following notes on the complete outfit of instruments and materials should be consulted before buying, because it is very essential to the best results that a good outfit be secured.

The complete outfit for students in mechanical drawing in Sibley College is as follows:

(1) THE DRAWING-BOARD for freshman work is 17″ × 22″ × $\frac{5}{8}$″, the same as that used for free-hand drawing. The board for sophomore and junior drawing is 20″ × 26″ × not more than $\frac{1}{2}$″ in thickness. The material should be soft pine and constructed as shown by Fig. 1.

(2) PAPER, quality and size to suit.

(3) PENCILS, one 6H and one 4H Koh-i-noor or Faber, also one Eagle Pilot No. 2 with rubber tip.

(4) The T-SQUARE for freshman work is furnished by the

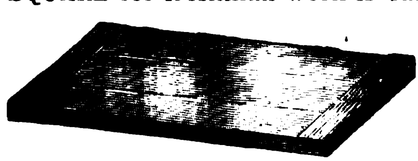

FIG. 1.

department; a plain pearwood T-square with a fixed head is all that is necessary for sophomore or junior work. Length to suit drawing-board.

(5) INSTRUMENTS. "The Sibley College Set," shown by Fig. 2, is recommended as a first-class medium-priced set of instruments. It contains:

FIG. 2.

A COMPASS, $5\frac{1}{2}''$ long, with fixed needle-point, pencil, pen and lengthening bar.

A SPRING BOW PENCIL, $3''$ long.

A " " PEN, $3''$ long.

A " " SPACER, $3''$ long.

A DRAWING-PEN, medium length.

A HAIR-SPRING DIVIDER, $5''$ long.

A nickel-plated box with leads.

(6) A TRIANGULAR BOXWOOD SCALE graduated as

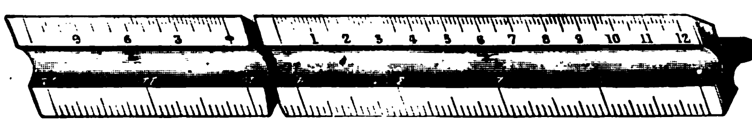

FIG. 3.

follows: $4''$ and $2''$, $3''$ and $1\frac{1}{2}''$, $1''$ and $\frac{1}{2}''$, $\frac{3}{4}''$ and $\frac{3}{8}''$, $\frac{1}{16}''$ and $\frac{1}{50}''$.

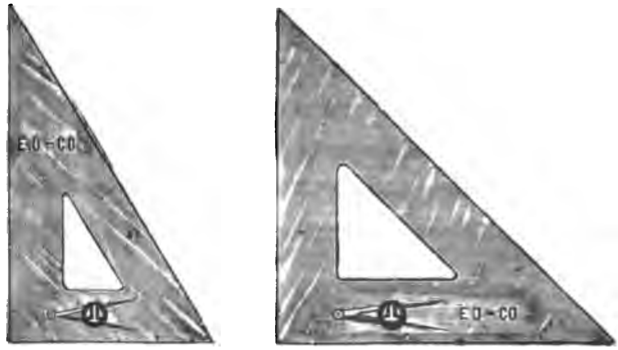

FIG. 4.

(7) 1 TRIANGLE $30^\circ \times 60^\circ$, celluloid, 10″ long.
1 " 45°, " 7″ "

(8) "SIBLEY COLLEGE SET" of IRREGULAR CURVES.

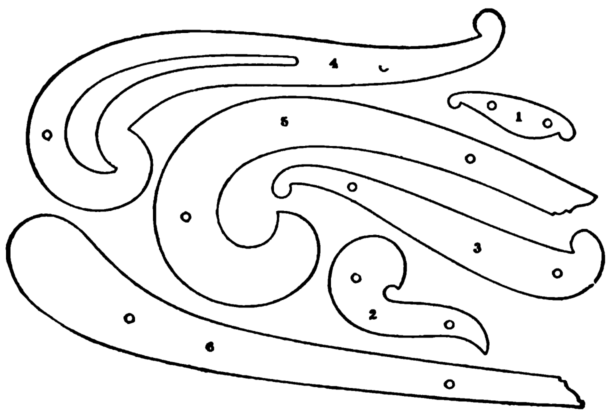

FIG. 5.

(9) GLASS-PAPER PENCIL SHARPENER.

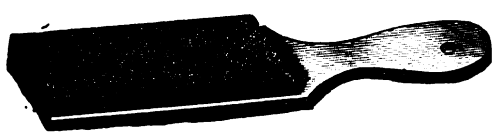

FIG. 6.

(10) INK, black waterproof, S. & H. Fig. 7.
(11) " red " Higgins. Fig. 8.
(12) " blue " "

FIG. 7.

FIG. 8.

(13) INK ERASER, Faber's Typewriter.

(14) PENCIL ERASER, Tower's Multiplex Rubber. Fig. 9.

(15) SPONGE RUBBER or FABER'S KNEADED RUBBER. Fig. 10.

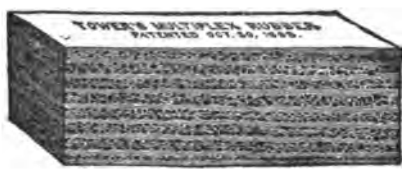

FIG. 9.

(16) TACKS, a small box of 1 oz. tacks.

(17) WATER-COLORS, $\frac{1}{2}$ pan each of Payne's Gray, Crimson Lake, Prussian Blue, Burnt Sienna, and Gamboge. Windsor & Newton. Fig. 11.

FIG. 10.

FIG. 11.

(18) TINTING BRUSH, Camel's Hair No. 10. Fig. 12.

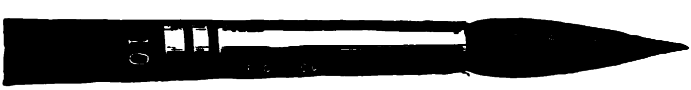

FIG. 12.

(19) TINTING SAUCER. Fig. 13.

(20) WATER GLASS. Fig. 14.

(21) ARKANSAS OIL-STONE. $2'' \times \frac{1}{2}'' \times \frac{1}{16}''$.

(22) PIECE OF SHEET CELLULOID, color No. 300, thickness $\frac{5}{1000}$, dull on both sides.

(23) PROTRACTOR, German silver, about 5″ diam. Fig. 15.

(24) SCALE GUARD, " " Fig. 16.

FIG. 13.

FIG. 14.

(25) SHEET OF TRACING-CLOTH, 18″ × 24″.

(26) WRITING-PEN, point, "Gillott" No. 303.

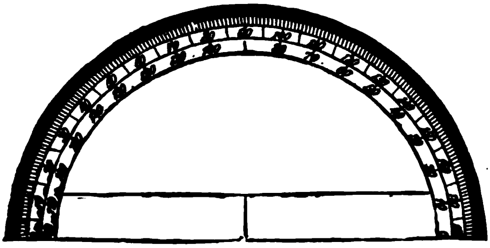

FIG. 15.

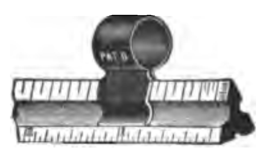

FIG. 16.

(27) Piece of SHEET BRASS, 4″×4″.

(28) NEEDLES, two with handles.

The following numbers of "The Complete Outfit" are all that the student will be required to purchase for freshman mechanical drawing (No. 2 Register, '97–'98): 2, 3, 5, 6, 7, 8, 9, 10, 13, 14, 16, 26.

The remainder of the outfit may be purchased during the sophomore and junior years.

INSTRUMENTS.

It is a common belief among students that any kind of cheap instrument will do with which to learn mechanical drawing, and not until they have acquired the proper use of the instruments should they spend money in buying a first-class set. This is one of the greatest mistakes that can be made. Many a student has been discouraged and disgusted because, try as he would, he could not make a good drawing, using a set of instruments with which it would be difficult for even an experienced draftsman to make a creditable showing.

If it is necessary to economize in this direction it is better and easier to get along with a fewer number, and have them of the best, than it is to have an elaborate outfit of questionable quality.

The instruments composing the "Sibley College Set" are made by T. Alteneder & Sons, and are certainly as good as the best. See Fig. 17.

USE OF INSTRUMENTS.

The Pencil.—Designs of all kinds are usually worked out in pencil first, and if to be finished and kept they are inked in and sometimes colored and shaded; but if the drawing is only to be finished in pencil, then all the lines except construction, center, and dimension lines should be made broad and dark,

so that the drawing will stand out clear and distinct. It will be noticed that this calls for two kinds of pencil-lines, the first a thin, even line made with a hard, fine-grained lead-pencil, not less than 6H (either Koh-i-noor or Faber's), and sharpened to a knife-edge in the following manner: The lead should be carefully bared of the wood with a knife for about $\frac{1}{8}''$, and the wood neatly tapered baçk from that point; then lay the lead upon the glass-paper sharpener illustrated in the outfit, and carefully rub to and fro until the pencil assumes a long taper from the wood to the point; now turn it over and do the same with the other side, using toward the last a slightly oscillating motion on both sides until the point has assumed a sharp, thin, knife-edge endwise and an elliptical contour the other way.

This point should then be polished on a piece of scrap drawing-paper until the rough burr left by the glass-paper is removed, leaving a smooth, keen, ideal pencil-point for drawing straight lines.

With such a point but little pressure is required in the hands of the draftsman to draw the most desirable line, one that can be easily erased when necessary and inked in to much better advantage than if the line had been made with a blunt point, because, when the pencil-point is blunt the inclination is to press hard upon it when drawing a line. This forms a groove in the paper which makes it very difficult to draw an even inked line.

The second kind of a pencil-line is the broad line, as explained above; it should be drawn with a somewhat softer pencil, say 4H, and a thicker point.

All lines not necessary to explain the drawing should be

erased before inking or broadening the pencil-lines, so as to make a minimum of erasing and cleaning after the drawing is finished.

When drawing pencil-lines, the pencil should be held in a plane passing through the edge of the T-square perpendicular to the plane of the paper and making an angle with the plane of the paper equal to about 60°.

Lines should always be drawn from left to right. A soft conical-pointed pencil should be used for lettering, figuring, and all free-hand work.

The Drawing-pen.—The best form, in the writer's opinion, is that shown in Fig. 17. The spring on the upper blade

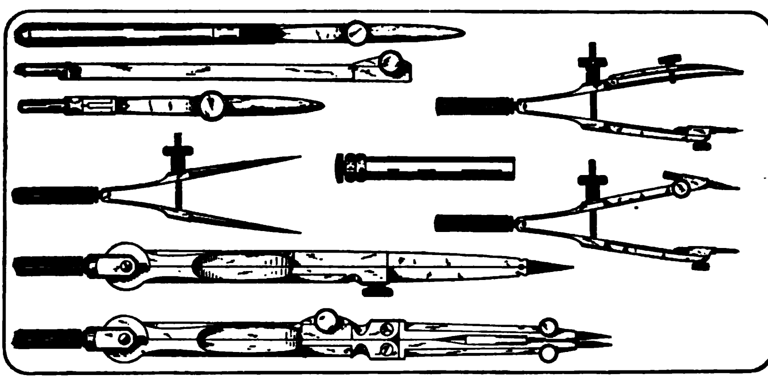

FIG. 17.

spreads the blades sufficiently apart to allow for thorough cleaning and sharpening. The hinged blade is therefore unnecessary. The pen should be held in a plane passing through the edge of the T-square at right angles to the plane of the paper, and making an angle with the plane of the paper ranging from 60° to 90°.

The best of drawing-pens will in time wear dull on the point, and until the student has learned from a competent

teacher how to sharpen his pens it would be better to have them sharpened by the manufacturer.

It is difficult to explain the method of sharpening a drawing-pen.

If one blade has worn shorter than the other, the blades should be brought together by means of the thumb-screw, and placing the pen in an upright position draw the point to and fro on the oil-stone in a plane perpendicular to it, raising and lowering the handle of the pen at the same time, to give the proper curve to the point. The Arkansas oil-stones (No. 21 of " The Complete Outfit ") are best for this purpose.

The blades should next be opened slightly, and holding the pen in the right hand in a nearly horizontal position, place the lower blade on the stone and move it quickly to and fro, slightly turning the pen with the fingers and elevating the handle a little at the end of each stroke. Having ground the lower blade a little, turn the pen completely over and grind the upper blade in a similar manner for about the same length of time; then clean the blades and examine the extreme points, and if there are still bright spots to be seen continue the grinding until they entirely disappear, and finish the sharpening by polishing on a piece of smooth leather.

The blades should not be too sharp, or they will cut the paper. The grinding should be continued only as long as the bright spots show on the points of the blades.

When inking, the pen should be held in about the same position as described for holding the pencil. Many draftsmen hold the pen vertically. The position may be varied with good results as the pen wears. Lines made with the pen should only be drawn from left to right.

THE TRIANGLES.

The triangles shown at Fig. 4 (in "The Complete Outfit") are 10″ and 7″ long respectively, and are made of transparent celluloid. The black rubber triangles sometimes used are but very little cheaper (about 10 cents) and soon become dirty when in use; the rubber is brittle and more easily broken than the celluloid.

Angles of 15°, 75°, 30°, 45°, 60°, and 90° can readily be drawn with the triangles and T-square. Lines parallel to oblique lines on the drawing can be drawn with the triangles by placing the edge representing the height of one of them so as to coincide with the given line, then place the edge representing the hypotenuse of the other against the corresponding edge of the first, and by sliding the upper on the lower when holding the lower firmly with the left hand any number of lines may be drawn parallel to the given line.

The methods of drawing perpendicular lines and making angles with other lines within the scope of the triangles and T-square are so evident that further explanation is unnecessary.

THE T-SQUARE.

The use of the T-square is very simple, and is accomplished by holding the head firmly with the left hand against the left-hand end of the drawing-board, leaving the right hand free to use the pen or pencil in drawing the required lines.

THE DRAWING-BOARD.

If the left-hand edge of the drawing-board is straight and

the T-square, then horizontal lines parallel to the upper edge of the paper and perpendicular to the left-hand edge may be drawn with the T-square, and lines perpendicular to these can be made by means of the triangles, or ***set squares***, as they are sometimes called.

THE SIBLEY COLLEGE SCALE.

This scale, illustrated in Fig. 3 (in "The Complete Outfit"), was arranged to suit the needs of the students in Sibley College. It is triangular and made of boxwood. The six edges are graduated as follows; $\frac{1}{16}''$ or full size, $\frac{1}{80}''$, $\frac{3}{4}''$ and $\frac{3}{8}'' =$ 1 ft., $1''$ and $\frac{1}{2}'' =$ 1 ft., $3''$ and $1\frac{1}{2}'' =$ 1 ft., and $4''$ and $2'' =$ 1 ft.

Drawings of very small objects are generally shown enlarged—e.g., if it is determined to make a drawing twice the full size of an object, then where the object measures one inch the drawing would be made $2''$, etc.

Larger objects or small machine parts are often drawn full size—i.e., the same size as the object really is—and the drawing is said to be made to the scale of full size.

Large machines and large details are usually made to a reduced scale—e.g., if a drawing is to be made to the scale of $2'' =$ 1 ft., then $2''$ measured by the standard rule would be divided into 12 equal parts and each part would represent $1''$.

THE SCALE GUARD.

This instrument is shown in Fig. 16 (in "The Complete Outfit"). It is employed to prevent the scale from turning, so that the draftsman can use it without having to look for

the particular edge he needs every time he wants to lay off a measurement.

THE COMPASSES.

When about to draw a circle or an arc of a circle, take hold of the compass at the joint with the thumb and two first fingers, guide the needle-point into the center and set the pencil or pen leg to the required radius, then move the thumb and forefinger up to the small handle provided at the top of the instrument, and beginning at the lowest point draw the line clockwise. The weight of the compass will be the only down pressure required.

The sharpening of the lead for the compasses is a very important matter, and cannot be emphasized too much. Before commencing a drawing it pays well to take time to properly sharpen the pencil and the lead for compasses and to keep them always in good condition.

The directions for sharpening the compass leads are the same as has already been given for the sharpening of the straight-line pencil.

THE DIVIDERS OR SPACERS.

This instrument should be held in the same manner as described for the compass. It is very useful in laying off equal distances on straight lines or circles. To divide a given line into any number of equal parts with the dividers, say 12, it is best to divide the line into three or four parts first, say 4, and then when one of these parts has been subdivided accurately into three equal parts, it will be a simple matter to step off these latter divisions on the remaining three-fourths

of the given line. Care should be taken not to make holes in the paper with the spacers, as it is difficult to ink over them without blotting.

THE SPRING BOWS.

These instruments are valuable for drawing the small circles and arcs of circles. It is very important that all the small arcs, such as fillets, round corners, etc., should be carefully pencilled in before beginning to ink a drawing. Many good drawings are spoiled because of the bad joints between small arcs and straight lines.

When commencing to ink a drawing, all small arcs and small circles should be inked first, then the larger arcs and circles, and the straight lines last. This is best, because it is much easier to know where to stop the arc line, and to draw the straight line tangent to it, than *vice versa.*

IRREGULAR CURVES.

The Sibley College Set of Irregular Curves shown in Fig. 5 are useful for drawing irregular curves through points that have already been found by construction, such as ellipses, cycloids, epicycloids, etc., as in the cases of gear-teeth, cam outlines, rotary pump wheels, etc.

When using these curves, that curve should be selected that will coincide with the greatest number of points on the line required.

THE PROTRACTOR.

This instrument is for measuring and constructing angles. It is shown in Fig. 15. It is used as follows when measuring

an angle: Place the lower straight edge on the straight line which forms one of the sides of the angle, with the nick exactly on the point of the angle to be measured. Then the number of degrees contained in the angle may be read from the left, clockwise.

In constructing an angle, place the nick at the point from which it is desired to draw the angle, and on the outer circumference of the protractor, find the figure corresponding to the number of degrees in the required angle, and mark a point on the paper as close as possible to the figure on the protractor; after removing the protractor, draw a line through this point to the nick, which will give the required angle.

SHADE LINES AND SHADING.

Shade Lines are quite generally used on engineering working drawings; they give a relieving appearance to the projecting parts, improve the looks of the drawing and make it easier to read, and are quickly and easily applied.

The *Shading* of the curved surfaces of machine parts is sometimes practiced on specially finished drawings, but on working drawings most employers will not allow shading because it takes too much time, and is not essential to a quick and correct reading of a drawing, especially if a system of shade lines is used.

The Source of Light is considered to be at an infinite distance from the object, therefore the Rays of Light will be represented by parallel lines.

The Source of Light is considered to be fixed, and the Point of Sight situated in front of the object and at an infinite dis-

tance from it, so that the *Visual Rays* are parallel to one another and per. to the plane of projection.

Shade Lines divide illuminated surfaces from dark surfaces.

Dark surfaces are not necessarily to be defined by those surfaces which are darkened by the shadow cast by another part of the object, but by reason of their location in relation to the rays of light.

It is the general practice to shade-line the different projections of an object as if each projection was in the same plane—e.g., suppose a cube, Fig. 18, situated in space in the third angle, the point of sight in front of it, and the direction

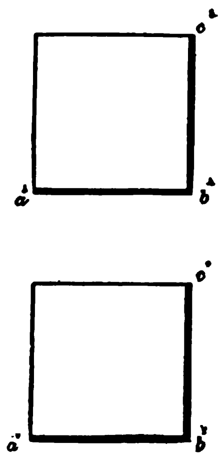

FIG. 18.

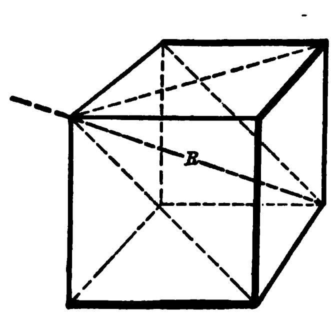

FIG. 19.

of the rays of light coinciding with the diagonal of the cube, as shown by Fig. 19. Then the edges $a^v b^v$, $b^v c^v$ will be shade lines, because they are the edges which separate the illuminated faces (the faces upon which fall the rays of light) from the shaded faces, as shown by Fig. 19.

Now the source of light being fixed, let the point of sight

remain in the same position, and conceive the object to be revolved through the angle of 90° about a hor. axis so that a plan at the top of the object is shown above the elevation, and as the projected rays of light falling in the direction of the diagonal of a cube make angles of 45° with the hor., then with the use of the 45° triangle we can easily determine that the lower and right-hand edges of the plan as well as of the elevation should be shade lines.

This practice then will be followed in this work, viz.:

Shade lines shall be applied to all *projections* of an object, considering the rays of light to fall upon each of them, from the same direction.

Shade lines should have a *width* equal to 3 times that of the other outlines. *Broken lines* should never be shade lines.

The outlines of *surfaces of revolution* should not be shade lines. The shade-lined figures which follow will assist in illustrating the above principles; they should be studied until understood.

WORKING DRAWINGS.

Working drawings are sometimes made on brown detail-paper in pencil, traced on tracing-paper or cloth, and then blue printed.

The latter process is accomplished as follows:

The tracing is placed face down on the glass in the printing-frame, and the prepared paper is placed behind it, with the sensitized surface in contact with the back of the tracing.

In printing from a negative the sensitized surface of the prepared paper is placed in contact with the film side of the negative, and the face is exposed to the light.

The blue-print system for working drawings has many drawbacks, e.g., the sectional parts of the drawing requires to be hatch-lined, using the standard conventions already referred to for the different materials. This takes a great deal of time. The print has usually to be mounted on cardboard, although this is not always done, and unless it is varnished the frequent handling with dirty, oily fingers soon makes it unfit for use.

Changes can be made on the prints with soda-water, it is true, but they seldom look well, and when many changes or additions require to be made it is best to make them on the tracing and take a new print. And the sunlight is not always favorable to quick printing. So taking everything into consideration the system of making working drawings directly on cards and varnishing them is probably the best. It is the system used by the Schenectady Locomotive Works and many other large engineering establishments. In size the cards are made 9″ × 12″, 12″ × 18″, 18″ × 24″; they are made of thick pasteboard mounted with Irish linen record-paper. The drawings are pencilled and inked on these cards in the usual way, and the sections are tinted with the conventional colors, which are much quicker applied than hatch-lines. The face of the drawing is protected with two coats of white shellac varnish, while the back of the card is usually given a coat of orange shellac.

The white varnish can easily be removed with a little alcohol, and changes made on the drawing, and when revarnished it is again ready for the shop.

In the hands of an experienced workman a working drawing is intended to convey to him all the necessary

information as to shape, size, material, and finish to enable him to properly construct it without any additional instructions. This means that it must have a sufficient number of elevations, sections, and plans to thoroughly explain and describe the object in every particular. And these views should be completely and conveniently dimensioned. The dimensions on the drawing must of course give the sizes to which the object is to be made, without reference to the scale to which it may be drawn. The title of a working drawing should be as brief as possible, and not very large—a neat, plain, free-hand printed letter is best for this purpose.

Finished parts are usually indicated by the letter "f," and if it is all to be finished, then below the title it is customary to write or print "finished all over."

The number of the drawing may be placed at the upper left-hand corner, and the *initials* of the draftsman immediately below it.

Lettering.—All lettering on mechanical drawings should be plain and legible, but the letters in a title or the figures on a drawing should never be so large as to make them appear more prominent than the drawing itself.

The best form of letter for practical use is that which gives the neatest appearance with a maximum of legibility and requires the least amount of time and labor in its construction.

Figuring.—Great care should be taken in figuring or dimensioning a mechanical drawing, and especially a working drawing.

To have a drawing accurately, legibly, and neatly figured is considered by practical men to be the most important part of a working drawing.

There should be absolutely no doubt whatever about the character of a number representing a dimension on a drawing.

Many mistakes have been made, incurring loss in time, labor, and money through a wrong reading of a dimension.

Drawings should be so fully dimensioned that there will be no need for the pattern-maker or machinist to measure any part of them. Indeed, means are taken to prevent him from doing so, because of the liability of the workman to make mistakes, so drawings are often made to scales which are difficult to measure with a common rule, such as 2″ and 4″ = 1 ft.

STANDARD CONVENTIONAL SECTION LINES.

Conventional section lines are placed on drawings to distinguish the different kinds of materials used when such drawings are to be finished in pencil, or traced for blue printing, or to be used for a reproduction of any kind.

Water-colors are nearly always used for finished drawings and sometimes for tracings and pencil drawings.

The color tints can be applied in much less time than it takes to hatch-line a drawing. So that the color method should be used whenever possible.

To apply the color tint.—Great care should be taken in determining the depth of the tint to be used; when only the section parts are to be colored the tints should be quite *light* because it is much easier to obtain an even wash and a softer and more artistic effect. Before applying the color the drawing board should be cleared of drawing instruments, etc., so that it may be easily turned to enable the student to keep

the bounding color line always to his left, and keeping the brush in such position that the color just touches the bounding line transfer the color to the drawing with long sweeps of the brush until the surface is covered. Press out all color remaining in the brush with the fingers and apply the brush again to the little puddles remaining on the paper. The brush will draw it back into itself and leave an even tint all over the section.

FIG. 20.—This figure shows a collection of hatch-lined sections that is now the almost universal practice among draftsmen in this and other countries, and may be considered standard.

No. 1. To the right is shown a section of a wall made of *rocks*. When used without color, as in tracing for printing, the rocks are simply shaded with India ink and a 175 Gillott steel pen. For a colored drawing the ground work is made of gamboge or burnt umber. To the left is the conventional representation of water for tracings. For colored drawings a blended wash of Prussian blue is added.

No. 2. *Convention for Marble.*—When colored, the whole section is made thoroughly wet and each stone is then streaked with Payne's gray.

No. 3. *Convention for Chestnut.*—When colored, a ground wash of gamboge with a little crimson lake and burnt umber is used. The colors for graining should be mixed in a separate dish, burnt umber with a little Payne's gray and crimson lake added in equal quantities and made dark enough to form a sufficient contrast to the ground color.

No. 4. *General Convention for Wood.*—When colored the ground work should be made with a light wash of burnt sienna.

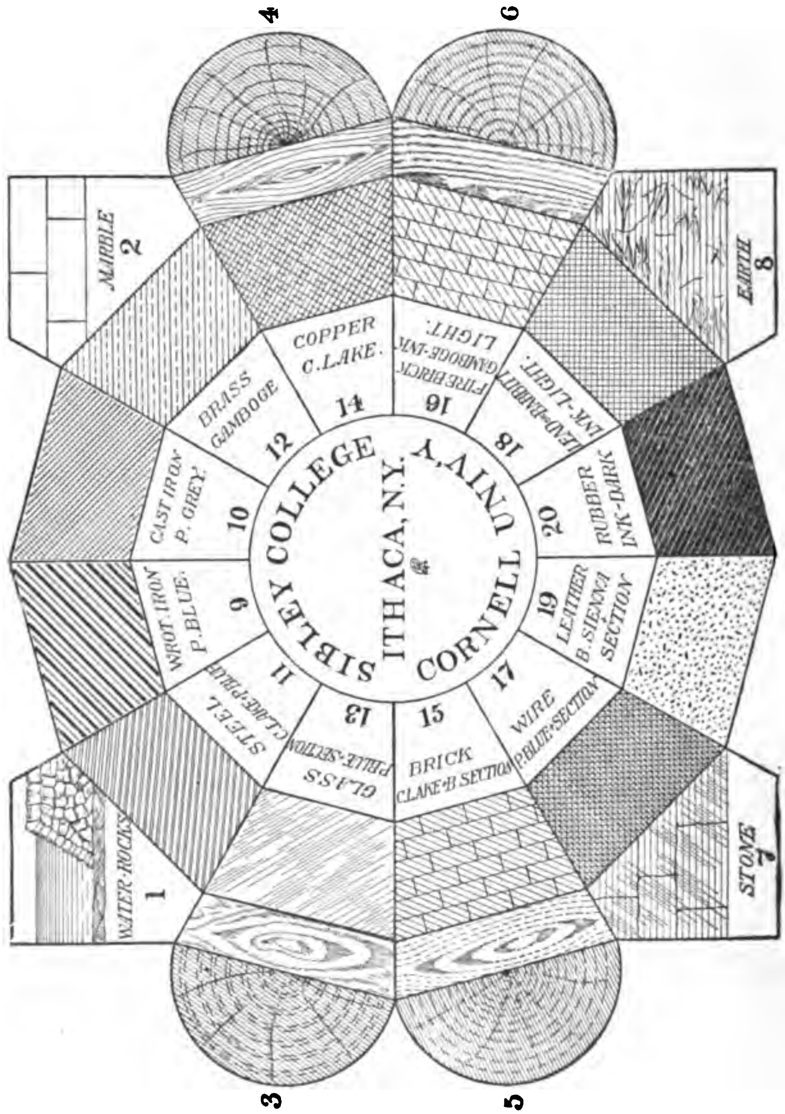
SIBLEY COLLEGE
ITHACA, N.Y.
CORNELL UNIV'Y
1
WATER-ROCKS
MARBLE
2
3
4
5
6
STONE
7
EARTH
8
9
WROT. IRON
P. BLUE.
10
CAST IRON
P. GREY.
11
STEEL
12
BRASS
GAMBOGE
13
GLASS
14
COPPER
C. LAKE.
15
BRICK
16
FIRE BRICK
LIGHT.
17
WIRE
18
19
LEATHER
B. SIENNA
SECTION
20
RUBBER
INK-DARK

FIG. 20.

The graining should be done with a writing-pen and a dark mixture of burnt sienna and a modicum of India ink.

No. 5. *Convention for Black Walnut.*—A mixture of Payne's gray, burnt umber and crimson lake in equal quantities is used for the ground color. The same mixture is used for graining when made dark by adding more burnt umber.

No. 6. *Convention for Hard Pine.*—For the ground color make a light wash of crimson lake, burnt umbeı, and gamboge, equal parts. For graining use a darker mixture of of crimson lake and burnt umber.

No. 7. *Convention for Building-stone.*—The ground color is a light wash of Payne's gray and the shade lines are added mechanically with the drawing-pen or free-hand with the writing-pen.

No. 8. *Convention for Earth.*—Ground color, India ink and neutral tint. The irregular lines to be added with a writing-pen and India ink.

No. 9. *Section Lining for Wrought or Malleable Iron.*—When the drawing is to be tinted, the color used is Prussian blue.

No. 10. *Cast Iron.*—These section lines should be drawn equidistant, not very far apart and narrower than the body lines of the drawing. The tint is Payne's gray.

No. 11. *Steel.*—This section is used for all kinds of steel. The lines should be of the same width as those used for cast-iron and the spaces between the double and single lines should be uniform. The color tint is Prussian blue with enough crimson lake added to make a warm purple.

No. 12. *Brass.*—This section is generally used for all kinds of composition brass, such as gun-metal, yellow metal,

bronze metal, Muntz metal, etc. The width of the full lines, dash lines and spaces should all be uniform. The color tint is a light wash of gamboge.

Nos. 13–20.—The section lines and color tints for these numbers are so plainly given in the figure that further instruction would seem to be superfluous.

Sometimes draftsmen will crosshatch all the sectional parts with a uniform space and ilne like that used for cast iron and mark the names of the different materials or their initials in some convenient place on the parts themselves. This does not look as well nor is it any more convenient to experienced men than the other method.

CONVENTIONAL LINES.

FIG. 21.—There are four kinds:

(1) *The Hidden Line.*—This line should be made of short dashes of uniform length and width, both depending somewhat on the size of the drawing. The width should always be slightly less than the body lines of the drawing, and the

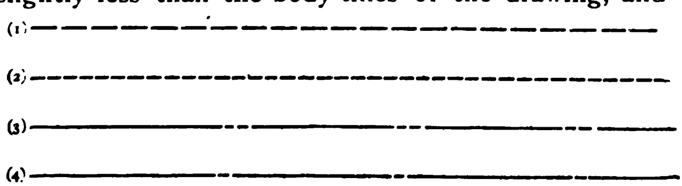

FIG. 21.

length of the dash should never exceed $\frac{1}{8}''$. The spaces between the dashes should all be uniform, quite small, never exceeding $\frac{1}{16}''$. This line is always inked in with *black ink.*

(2) *The Line of Motion.*—This line is used to indicate *point paths.* The dashes should be made shorter than those of the *hidden line,* just a trifle longer than dots. The spaces should of course be short and uniform.

(3) *Center Lines.*—Most drawings of machines and parts of machines are symmetrical about their center lines. When penciling a drawing these lines may be drawn continuous and as fine as possible, but on drawings for reproductions the black-inked line should be a long narrow dash and two short ones alternately. When colored inks are used the center line should be made a continuous *red* line and as fine as it is possible to make it.

(4) *Dimension Lines and Line of Section.*—These lines are made in *black* with a fine long dash and one short dash alternately. In color they should be continuous *blue* lines. Colored lines should be used wherever feasible, because they are so quickly drawn and when made fine they give the drawing a much neater appearance than when the conventional black lines are used. Colored lines should *never* be broken.

CONVENTIONAL BREAKS.

FIG. 22.—Breaks are used in drawings sometimes to indicate that the thing is actually longer than it is drawn, some-

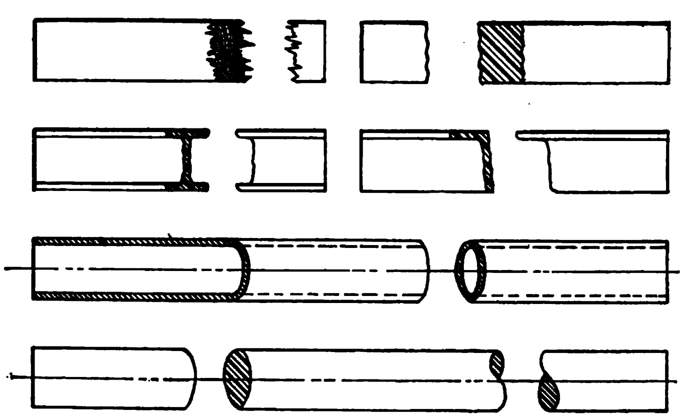

FIG. 22.

times to show the shape of the cross-section and the kind of material. Those given in Fig. 22 show the usual practice.

CROSS-SECTIONS.

FIG. 23.—When a cross-section of a pulley, gear-wheel or other similar object is required and the cutting-plane passes through one of the spokes or arms, then only the rim and hub should be sectioned, as shown at *xx* No. 1 and *zz* No. 2, and the arm or spoke simply outlined. Cross-sections of the arms may be made as shown at *AA* No. 2. In working drawings of gear-wheels only the number of teeth included in one quadrant need be drawn; the balance is usually shown by conventional lines, e.g., the *pitch* line the same as a center line, viz., a long

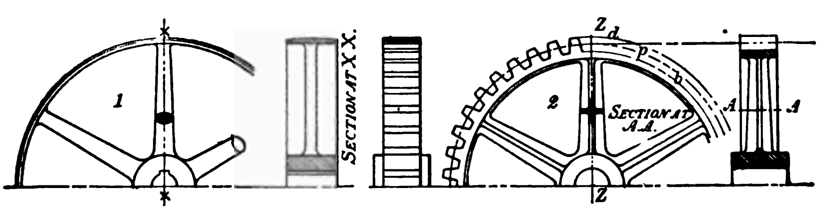

FIG. 23.

dash and two very short ones alternately or a fine continuous *red* line.

The *addendum* line (*d*) and the *root* or *bottom line* (*b*) the same as a dimension line, viz., one long dash and one short dash alternately or a fine continuous *blue* line. The end elevation of the gear-teeth should be made by projecting only the points of the teeth, as shown at No. 2.

Other conventions will be referred to in the text connected with the figures in which they are illustrated.

Constructions.—To draw the curve of intersection that is formed by a plane cutting an irregular surface of revolution.

Figs. 24 and 25 show examples of engine connecting-rod ends where the curve *I* is formed by the intersection of

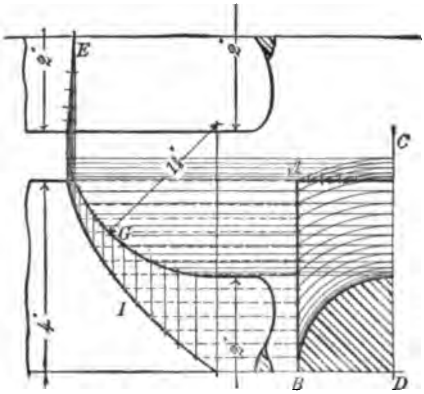

FIG. 24.

the flat stub end with the surface of revolution of the turned part of the rod.

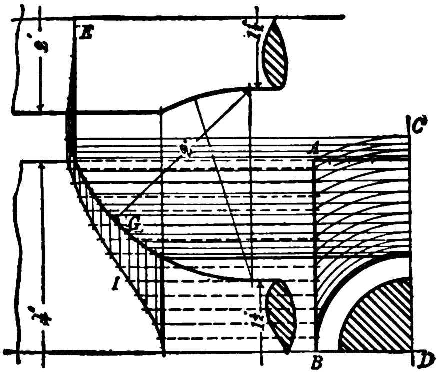

FIG. 25.

Divide the line *AB*, Figs. 24 and 25, into any number of equal parts and through them describe arcs cutting the center line *CD*. Through the intersections of these arcs with *CD* draw horizontals to intersect the curve or fillet *G*.

Through the intersections on G draw perpendiculars and from the divisions on AB draw horizontals to intersect the perpendiculars; these latter intersections are points in the curve I.

The curve E can be found in a similar way as shown by the figure.

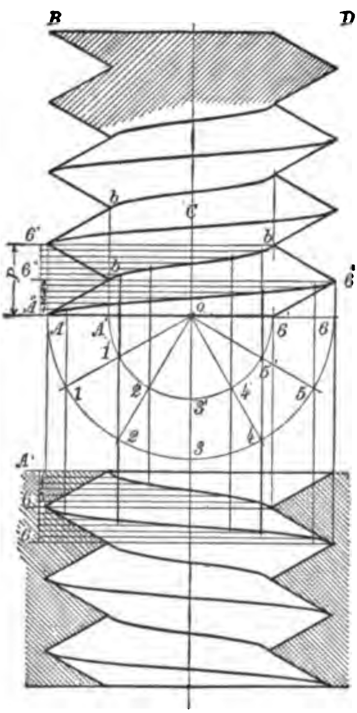

FIG. 26.

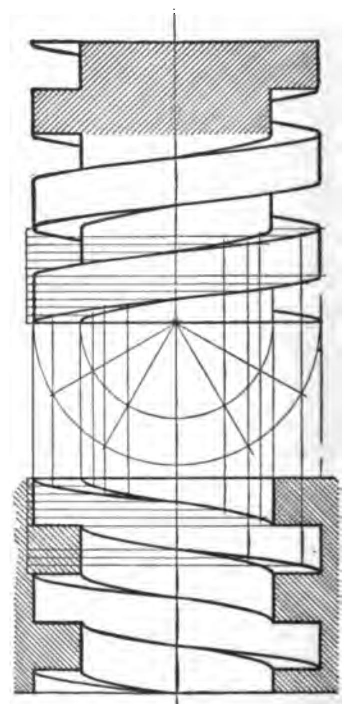

FIG. 27.

To draw the projections of a V-threaded screw and its nut of 3″ diam. and ¾″ pitch.

Begin by drawing the center line C, Fig. 26, and lay off on each side of it the radius of the screw 1½″. Draw AB and $6D$. Draw $A6$ the bottom of the screw, and on AB step off the pitch = ¾″, beginning at the point A.

On line $6D$ from the point 6 lay off a distance = half the pitch = $\frac{3}{8}''$, because when the point of the thread has completed half a revolution it will have risen perpendicularly a distance = half the pitch, viz., $\frac{3}{8}''$.

Then from the point $6''$ on $6D$ step off as many pitches as may be desired. From the points of the threads just found, draw with the 30° triangle and T-square the V of the threads intersecting at the points b . . b . . the bottom of the threads.

At the point O on line $A6$ draw two semicircles with radii ‖ the top and bottom of the thread respectively. Divide these into any number of equal parts and also the pitch P into the same number of equal parts. Through these divisions draw hors. and pers. intersecting each other in the points as shown by Fig. 26, which shows an elevation partly in section and a section of a nut to fit the screw. Through the points of intersection draw the curves of the helices shown, using No. 3 of the "Sibley College Set" of Irregular Curves.

ELEMENTARY MACHINE DESIGN.

A machine, according to Prof. John H. Barr, is "a combination of resistant bodies for modifying energy and doing work, the members of which are so arranged that, in operation, the motion of any member involves definite, relative, constrained motion of the others."

In order to obtain the most desirable results in designing such a structure it is necessary to give the several bodies composing it such form and proportion as will enable them to perform their functions in the best possible way and at the same time present a pleasing appearance to the experienced

eye. And, moreover, it must not be forgotten that these desired results should be sought with a due regard to *economy* of material and construction.

The *form* of a machine will probably depend largely upon the designer's experience and his natural ability or intuition.

The proportion of the several parts may be calculated if the opposing forces are known, but in many cases these forces cannot be accurately determined and the designer must rely upon the most approved practice of the past had under similar conditions.

MATERIALS USED IN MACHINE CONSTRUCTION.

The principal materials used in machine construction may be divided into three heads, viz.: Cast Metals, Wrought Metals, and Wood.

CAST METALS.

Among the cast metals the more important in machine construction are cast iron, malleable cast iron, cast steel, brass, copper-bronze or gun-metal, phosphor-bronze, and aluminum.

Cast Iron.—Three kinds of white cast iron and three of gray are used in different ways in machine construction. The whitest iron is very hard and is used like the others of its class for making wrought iron.

The gray irons do not melt as readily as the white, but are more fluid when melted. The grayest irons are the weakest and are used only for mixing with others in the cupola.

Ordinary cast iron contains from 3% to 5% of carbon, which in the white iron is fully combined with the iron, while only .6% to 1.5% is combined in the gray iron and 2.9% to 3.7% shows as graphite crystals.

Iron castings of machine-parts are made from patterns. These patterns are made of wood, usually soft pine, in form exactly like the castings desired. The patterns are used to make moulds in sand in the foundry and into these moulds is poured the molten iron.

Cast iron after solidifying in the moulds contracts while cooling about $\frac{1}{8}''$ per foot of length. To allow for this contraction pattern-makers use a special rule called a shrink-rule for measuring patterns; it is $\frac{1}{8}''$ per foot longer than the standard rule.

Sharp corners in patterns do not cast sharp and square in the metal, but come out ragged and blunt, so that whenever possible *sharp edges should be rounded* and *sharp concave corners filleted or partially filled in;* the result is a stronger and better-looking casting.

To avoid irregular internal strains in iron castings when cooling it is necessary that the section of the casting be made as uniform as possible, so that the metal may contract uniformly throughout.

Chilled Castings.—Melted gray cast iron if cooled quickly retains in chemical combination a large amount of carbon which otherwise would be separated from the casting. The result is a white hard iron called chilled cast iron. To secure this quick cooling the mould into which the metal is cast is made of thick cast iron, which draws the heat from the molten metal in much less time than does the sand mould.

Malleable Castings are made by putting a gray-iron casting in a suitable box and covering it with powdered red hematite, which is an oxide of iron, and keeping it in a furnace at a bright-red heat for from two to thirty hours or even longer, depending upon the size of the casting; such castings are valuable for small light parts of machines, because they are tough and strong. Malleable castings can be worked like wrought iron, but will not weld.

Cast Steel is made by melting broken pieces of blister-steel in a closed crucible and casting into ingots.

Brass is very much used, because it is easy to work, is cheap, strong, and tough, and of a good color. The usual composition of brass is 2 of copper to 1 of zinc, with sometimes a little lead added.

Muntz Metal is a brass composition of 3 parts copper to 2 of zinc. It can be rolled or forged when hot and is used in the shape of bolts and nuts, sheets for sheathing wooden vessels, and often takes the place of iron or steel because of its ability to withstand the corrosive action of water.

Copper.—Pure copper with a small addition of phosphorus makes fairly good castings, but it is difficult to obtain sound castings from copper alone. Copper has a reddish-brown color and is very malleable and ductile when pure. It can be hammered, rolled, and forged when hot or cold; joints can be united by brazing, but welding is difficult. The annealing of iron and steel is effected by heating and slow cooling, while copper can only be annealed by heating and quick cooling.

Bronze or Gun-metal.—The best composition is made of 9 parts of copper to 1 of tin. For bearings designed to sus-

tain great pressure very hard bronze is often used, in which the proportion of tin is increased to 14 parts with 86 parts of copper.

Phosphor-bronze.—This alloy is made by adding from 2% to 4% of phosphorus to the common bronze. It is used for many things in place of iron and steel, such as pump-rods, ship-propellers, etc.; it is also used quite largely for locomotive axle-bearings and shows excellent wearing qualities.

Babbitt Metal.—This is a soft white metal that is used quite largely for lining shaft-bearings. Its composition is usually as follows: copper 4 parts, antimony 8, tin 24, melted together, and before using this alloy is melted with an addition of twice its weight of tin and applied to the bearings while molten. So the real composition of the lining is copper 4, antimony 8, and tin 96.

Aluminum.—This is a very light metal, soft, malleable, and ductile, and of a silvery-white color with a bluish tint. A process for producing it with comparative cheapness was discovered in 1890, and since then its production has been rapidly increasing. It is thoroughly non-corrosive.

WROUGHT METALS.

These consist of wrought iron and steel of various qualities.

Wrought Iron or Malleable Iron is a white metal not easily melted and is very strong and tough. It is made from the white cast irons by abstracting the most of the latter's carbon in a puddling-furnace. It is taken from this furnace in large spongy masses called blooms, and shingled by repeated squeezing and hammering and rolled into what is known as puddled bars. The puddled bars are then cut into short

pieces and piled into faggots; these are heated again and rolled into what is known as merchant bars. The best qualities of wrought iron are piled together, reheated, and rolled in the same way many times, giving the iron its fibrous nature which makes it so tough and strong. A valuable property of wrought iron is that it can be welded at a temperature of from 1500° to 1600° Fahr.

Case-hardening.—This is a hardening of the surface of finished parts of machines, such as the links, guides, etc., of steam-engines, so that their wearing qualities are very much increased. It is effected as follows: the piece to be case-hardened is placed in a suitable receptacle and surrounded by bone-dust, horn-shavings, yellow prussiate of potash, or any such substance that is rich in carbon, and heated to about a red heat, when the wrought iron will absorb some of the carbon surrounding it and be converted into steel, which can be hardened by immersing in water.

Steel is made from wrought iron by adding a little carbon or from cast iron by extracting some of its carbon. There are three ways of doing this: the Bessemer, Siemens-Martin, and cementation processes.

Bessemer Steel is made by pouring melted cast iron into a converter through which a blast of air is forced. In this way the carbon in the cast iron is burnt out, leaving almost pure iron. To this is added a certain quantity of spiegeleisen, which is a compound of iron, carbon, and manganese, and then the molten metal is cast into steel ingots.

Siemens-Martin Steel is made by melting wrought iron and cast iron, or cast iron and certain kinds of iron ore, together on the hearth of a reverberatory gas-furnace.

The *Cementation Process* consists of embedding bars of wrought iron in powdered charcoal in a fire-clay trough and placed in a furnace for several days at a high temperature. The iron combines with portions of the carbon and forms blister-steel, so called from the blisters found on its surface. Bars of blister-steel about 18″ long are then bound together by strong steel wire and heated to a welding heat, then hammered and rolled into bars called shear-steel.

WOODS.

The woods used in machine construction are principally pine, fir, beech, boxwood, ash, elm, hornbeam, lignum-vitæ, mahogany, oak, and teak.

Pine and Fir are strong, cheap, and easy to work, and are largely used for a variety of purposes.

White and *Yellow Pine* are much used in pattern-making.

Beech is used for the cogs of mortise-wheels; it takes a smooth surface and is very close-grained.

Boxwood is much used for sheaves of pulley-blocks and bearings. It takes a smooth surface, is hard, heavy, and of a bright-yellow color.

Elm is very durable in water, and is therefore used for paddle-wheel floats, piles, etc.

Hornbeam is often used for cogs of mortise-wheels.

Lignum-vitæ.—This is a very hard wood of great strength and durability under water. For these reasons it is used for bearings under water and other purposes requiring hardness and strength. Its specific gravity is 1.33; i.e., $1\frac{1}{3}$ times the weight of the same volume of water.

Mahogany is a favorite for making small patterns. It is

straight-grained, strong, and durable, and does not as readily change its form when seasoning as most other woods.

Oak is tough and straight-grained, very durable, whether used dry or in water. It is used for machine-framing and supports.

Teak is a strong, tough, durable wood. It shrinks very little when seasoning, and is very valuable on that account. Bolts passing through it are prevented from rusting by the oil it contains.

STRENGTH OF MATERIALS.

DEFINITIONS.

Load.—The load on any member of a machine is a total of the external forces acting on it. The useful load is the load which the member is designed to carry outside of itself; e.g., the useful load on the springs of a railway-car is the load which may be placed upon the car in addition to the load arising from the weight of the car itself. A live load is a variable load applied and removed continuously. A dead load or constant load is that which has an unvarying and continuous straining action.

Strain and Stress.—Strain is the change of form produced by the action of a load. If the load does not exceed the elastic limit of the material the strain will disappear when the load is removed. Machine-members should be designed strong enough to resist permanent set under maximum load.

Stress is the force which causes strain. The different kinds of stress are: *tensile* stress or pull, *compressive* stress or

thrust, *shearing* stress or cross-cutting, *bending* or combined thrust and pull, and *torsional* or twisting stress.

Resistance of metal to change of form is due to the inherent cohesive force of its molecules.

Elasticity or spring is the inherent property in a material of regaining original form after an external load has been removed.

Elastic Limit.—The elastic limit is the limit of extension or compression to which a material can be subjected without permanent set. Within the elastic limit strain and stress are proportional.

Modulus of Elasticity. — Dr. Thomas Young of the British Royal Society propounded the following formula for the modulus of elasticity (E) in 1826, known as "Young's Modulus":

$$E = \frac{\text{stress per sq. in. in lbs.}}{\text{strain per inch of length}} \text{ (within the elastic limit).}$$

TABLE 1.

ELASTIC MODULI.

Material.	Modulus of Elasticity.	Material.	Modulus of Elasticity.
Wrought iron, bars....	29,000,000	Pine, average	1,500,000
Wrought iron, plates..	26,000,000	Beech	1,350,000
Cast steel	30,000,000	Boxwood	1,800,000
Cast steel, tempered...	36,000,000	Ash	1,600,000
Forged steel...........	30,000,000	Elm	900,000
Steel plates	31,000,000	Lignum-vitæ	1,000,000
Cast iron, average.....	17,000,000	Mahogany	1,300,000
Brass and bronze......	12,000,000	Oak, English	1,700,000
Muntz metal...........	14,000,000	Teak	2,300,000
Copper, average.	12,000,000	Leather	24,500
Lead, sheet............	720,000	Glass, plate	8,000,000

Ultimate Strength is the smallest load that will fracture a member under stress.

The Proof Strength is nearly equal to the load that will cause permanent set; i.e., to the maximum elastic resistance.

The Factor of Safety is the ratio of the ultimate strength of a member to the working load, or the breaking load to the actual load. The factor of safety changes for different materials and for different uses of the same material. It is of course much greater under live loads than under constant dead loads.

The following table gives the ordinary factors of safety in general use:

TABLE 2.

FACTORS OF SAFETY.

Material.	Ratio of Ultimate Load to Working Load.		
	Dead Load.	Live Load.	Shocks.
Cast iron	4	7	15
Wrought iron	3	5 to 8	9 to 13
Mild steel	3	5 to 8	9 to 13
Cast steel	3	5 to 8	10 to 15
Copper and similar metals and alloys	5	8	10 to 15
Wood	8	10	14 to 18
Brick and stone	10	20	30

Strength of Cast Iron.—The average American cast iron has a tenacity of about 20,000 lbs. per sq. in., but cast iron has been made which showed an ultimate tensile strength of 35,000 lbs. per sq. in.

The ultimate *compressive* strength of cast iron is from 4 to 6 times its tenacity,—the average is about 90,000 lbs. per sq. in.,—and the average *shearing* strength is about 20,000 lbs. per sq. in. The elastic limit of cast iron is from $\frac{3}{8}$ to nearly equal to the breaking strength.

Strength of Wrought Iron.—The elastic strength of wrought iron is usually over half its ultimate strength; good bars and plates will show an elastic limit of about 26,000 lbs. In ascertaining the strength of a particular piece of wrought iron it will be necessary to know the elongation per cent of specimen. The elongation is greater for short than for long specimens. The usual length of specimens for tensile test is 8″. Wrought iron loses its strength in forging; this loss of strength is equal to about 20%. The difference between the strength of wrought iron when pulled against the grain and in the direction of the grain is from 3000 to 9000 lbs. per sq. in., the strength in the direction of the grain being the greater. The tensile strength of wrought iron varies from 40,000 to 60,000 lbs. per sq. in.

Strength of Steel.—The steel cast from blister-steel is the strongest, having a tensile strength of from 100,000 to 130,000 lbs. per sq. in., but it is hard and brittle, with an elongation of only about 5%. It is therefore unsuitable for constructive purposes. A good plate steel for steam-boilers has a tensile strength of from 55,000 to 60,000 lbs., with an elongation of about 20% in a length of 8″.

The following tables were compiled after consulting various authorities; e.g., Thurston, Unwin, Kent, Molesworth, etc.

TABLE 3.

Average Ultimate and Elastic Strength of Various Materials and Moduli of Elasticity in Pounds per Square Inch.

Material.	Ultimate Strength.			Elastic Strength.			E. With the Grain.	E'. Transverse.
	Tension.	Compression.	Shearing.	Tension.	Compression.	Shearing.		
Cast iron, common...	20,000	90,000	20,000	12,000	23,000	9,000	15,000,000	7,000,000
Wrought iron, bars..	50,000	48,000	40,000	26,000	26,000	22,000	29,000,000	16,500,000
Wrought iron, plates.	48,000	46,000	38,000	26,000	26,000	22,000	29,000,000	10,000,000
Wrought shape iron..	48,000	46,000	38,000	26,000	26,000	22,000	29,000,000	10,000,000
Wrought stay-bolt iron	51,000	49,000	40,000	28,000	28,000	24,000	29,000,000	10,000,000
Wrought rivets	50,000	48,000	38,000	26,000	26,000	22,000	29,000,000	10,000,000
Malleable cast iron...	32,000	....	38,000	23,500	27,000	20,000	24,500,000	13,500,000
Cast steel............	88,000	125,000	64,000	70,000	....	64,000	30,000,000	11,000,000
Soft-steel plates......	55,000	66,000	50,000	32,000	....	25,000	30,000,000	11,000,000
Steel rivets........	52,665	50,000	45,000	29,000	....	....	29,000,000	10,500,000
Cast copper..........	23,000	45,000	....	5,900	....	....	....	....
Forged copper	34,000	58,000	....	4,500	4,000	3,000	15,000,000	6,000,000
Brass, yellow........	17,500	10,500	....	7,000	....	5,200	9,200,000	3,500,000
Gun-metal	36,000	....	....	6,200	....	4,200	10,000,000	4,000,000
Wood, pine	12,000	6,000	650	.....	....	....	1,600,000	90,000
Wood, oak, English .	15,000	10,000	2,300	....	....	......	1,700,000	82,000
Leather.........	4,200	....	....	....	....	....	25,000	....

Wrought Iron has a specific gravity of 7.5 to 7.8 according to its chemical composition and physical structure.

Cast Iron has a specific gravity of 7.25.

The tensile strength of metals varies with their temperature, generally decreasing as their temperature is increased.

TABLE 4.

Relative Tenacities of Metal. (Thurston.)

Lead	1.0	Cast iron	7 to 12
Tin	1.3	Wrought iron . .	20 to 40
Zinc	2.0	Steel	40 to 100
Worked copper . .	12 to 20		

USEFUL TABLES AND MISCELLANEOUS INFORMATION.

WEIGHTS AND MEASURES.

Avoirdupois or Commercial Weight.

16 drachms	1 ounce.
16 ounces	1 pound.
14 pounds	1 stone.
28 "	1 quarter.
4 quarters	1 cwt.
2240 pounds	1 ton.

Square Measure.

144	square inches	1	square	foot.
9	" feet	1	"	yard.
30¼	" yards	1	"	rod.
40	" rods	1	"	rood.
4	" roods	1	"	acre.
640	" acres	1	"	mile.

Measure of Volume.

A cubic foot has	1728 cubic inches.
An ale gallon has	282 " "
A standard or wine gallon has	231 " "
A dry gallon has	268.8 " "
A bushel has	2150.4 " "
A cord of wood has	128 " feet.
A perch of stone has	24.75 " "
A ton of round timber has	40 " "
A " hewn " "	50 " "
A box 19⅜ × 19⅜ inches, 19⅜ inches deep, contains	1 barrel.
A " 12 15/16 × 12 15/16 " 12 15/16 " " "	1 bushel.
A " 8⅛ × 8⅛ " 8⅛ " " "	1 peck.
A " 6 7/16 × 6 7/16 " 6 7/16 " " "	½ "
A " 4 1/16 × 4 1/16 " 4 1/16 " " "	1 quart.
An acre contains	4840 square yards.
209 feet long by 209 feet broad is	1 acre.

Table of Distance.

A mile is	5280 feet or 1760 yards.
A knot is	6086 feet.
A league is	3 miles.
A fathom is	6 feet.
A metre is nearly	3 feet 3⅜ inches.
A hand is	4 inches.
A palm is	3 "
A span is	9 "

Measure of Length.

12 inches	1 foot.
3 feet	1 yard.
2 yards	1 fathom.
16½ feet	1 rod.
4 rods	1 chain.
10 chains	1 furlong.
8 furlongs	1 mile.
3 miles	1 league.

Each *nominal* horse-power of boilers requires 1 cubic foot of water per hour.

In calculating horse-power of steam-boilers consider for—
Tubular boilers 15 sq. ft. of heating-surface equivalent to 1 horse-power.
Flue boilers 12 sq. ft. of heating-surface equivalent to 1 horse-power.
Cylinder boilers 10 sq. ft. of heating-surface equivalent to 1 horse-power.

To find the area of a piston, square the diameter and multiply by .7854.

To find the pressure in pounds per square inch of a column of water, multiply the height of the column in feet by .434.

A horse-power in machinery is estimated at 33,000 pounds raised one foot high in a minute, or one pound raised 33,000 feet high in a minute.

Iron under the influence of the hammer and of constant use gradually assumes, by repeated vibration, a different texture from that it had when the piece was new. The metal becomes crystalline, loses its tenacity, and becomes brittle.

WEIGHT OF WATER.

One cubic foot at 39.1° F. = 62.425 lbs., at 212° F. = 59.833. At 62° F. the weight varies from 62.291 to 62.360. The figure generally believed to be the most accurate is 62.355. Weight of 1 gallon at 39.2° = 8.3389 lbs.

WEIGHTS OF CAST-IRON WATER-PIPES.

IN POUNDS PER FOOT RUN, INCLUDING BELLS AND SPIGOTS.

Diameter.	Philadelphia Water-works.	Chicago Water-works.	Cincinnati.		Regular Standard.	Light.
			Weight.	Thickness.		
2-inch....	....	....	....	....	7	6
3 "	15.000	....	17	½"	15	13
4 " ...	21.111	24.167	23	"	22	20
6 "	30.106	36.666	50	¾"	33	30
8 "	40.683	50.000	65	"	42	40
10 "	52.075	65.000	80	"	60	55
12 "	69.162	83.333	100	"	75	70
16 "	102.522	125.000	130	"	....	....
20 "	147.681	....	200	⅞"	....	....
24 "	....	250.000	224	"	....	....
30 "	....	....	300	1"	....	....
36 "	....	450.000	430	1⅛"	....	....

Water-pipe is usually tested to 300 pounds pressure per square inch before delivery, and a hammer test should be made while the pipe is under pressure.

The Cincinnati lengths are uniform for all diameters, 12 feet exclusive of bell.

Standard lengths are for 2-inch pipe 8 feet, and all other sizes 12 feet.

THICKNESS OF CAST-IRON WATER-PIPE.

The following formula, adapted from Neville, is believed to be a safe equation for the thickness of cast-iron pipe for public water-supply:

$$t = \frac{9}{S}\left[.0016\left(\frac{h}{33} + 10\right)d\right] + .32,$$

where t = thickness of pipe in inches,
h = head or pressure in feet,
d = diameter of pipe in inches,
S = the tensile strength of metal in tons of 2000 pounds.

What should be the thickness of a 20-inch water-main subject to a maximum pressure of 150 pounds per square inch, or $150 \times 2.308 = 346.2$ feet head, with cast-iron of 18,000 pounds tensile strength?

$$t = \frac{9}{9} \times \left[.0016\left(\frac{346.2}{33} + 10\right) \times 20\right] + .32 = .9757''.$$

What should be the thickness of 40-inch pipe for same service and of same metal?

$$t = \frac{9}{9} \times \left[.0016\left(\frac{346.2}{33} + 10\right) \times 40\right] + .32 = 1.6313''$$

The speed at which millstones should be run is

For 3-feet stones	230 to 250	revolutions per minute.
" 3½ " "	200	" " "
" 4 " "	180	" " "
" 4½ " "	160	" " "
Speed of bolting-reels	30 to 35	" " "
" " conveyers for flour	35 to 40	" " "
" " " " wheat	45 to 50	" " "
" " elevators	30 to 35	" " "

" " smut-machines from 550 to 700 revolutions per minute, according to size of machine.

For *merchant* mills allow 20 horse-power to a pair of burrs (4 feet), and the necessary machinery for cleaning and bolting; and for *country* mills about 10 horse-power to a pair of burrs.

For a single upright saw allow 10 horse-power, speed about 150 revolutions per minute.

For circular saws the best average working speed is

650 to 700 rev. per min. for 36-in. saw.	500 to 525 rev. per min. for 48-in. saw.
600 to 650 " " " " 40 " "	475 to 500 " " " " 54 " "
550 to 600 " " " " 42 " "	400 to 450 " " " " 60 " "
525 to 550 " " " " 44 " "	

A 60-saw gin requires 6 horse-power to gin 500 pounds of lint in 2 hours.

A sumac-mill requires 15 horse-power.

To reduce for round cores and core-prints, multiply the square of the diameter by the length of the core in inches, and the product by 0.017 is the weight of the pine core, to be deducted for the weight of the pattern.

SHRINKAGE OF CASTINGS.

Pattern-maker's rule should be for	Cast iron	1/8	of an inch longer per linear foot.
	Brass	3/16	
	Lead	1/8	
	Tin	1/12	
	Zinc	3/16	

PROPERTIES OF THE CIRCLE.

Diameter × 3.14159 = circumference.
Diameter × .8862 = side of an equal square.
Diameter × .7071 = side of an inscribed square.
Diameter^2 × .7854 = area of circle.
Radius × 6.28318 = circumference.
Circumference ÷ 3.14159 = diameter.

WROUGHT-IRON WELDED TUBES FOR STEAM, GAS, OR WATER.

Nominal Diameter.	Actual Inside Diameter.	Actual Outside Diameter.	Thickness.	Weight per Foot of Length.	No. of Threads per Inch of Screw.
Inches.	Inches.	Inches.	Inches.	Pounds.	
1/8	.270	.405	.068	.243	27
1/4	.364	.54	.088	.422	18
3/8	.494	.675	.091	.561	18
1/2	.623	.84	.109	.845	14
3/4	.824	1.05	.113	1.126	14
1	1.048	1.315	.134	1.670	11½
1¼	1.380	1.66	.140	2.258	11½
1½	1.611	1.9	.145	2.694	11½
2	2.067	2.375	.154	3.667	11½
2½	2.468	2.875	.204	5.773	8
3	3.067	3.5	.217	7.547	8
3½	3.548	4.0	.226	9.055	8
4	4.026	4.5	.237	10.728	8
4½	4.508	5.0	.247	12.492	8
5	5.045	5.563	.259	14.564	8
6	6.065	6.625	.280	18.767	8
7	7.023	7.625	.301	23.410	8
8	7.982	8.625	.322	28.348	8
9	9.001	9.688	.344	34.077	8
10	10.019	10.75	.366	40.641	8

DIFFERENT COLORS OF IRON CAUSED BY HEAT. (POUILLET.)

Cent.	Fahr.	Color.
210°	410°	Pale yellow.
221	430	Dull yellow.
256	493	Crimson.
261 370	502 680	Violet, purple, and dull blue; between 261° C. and 370° C. it passes to bright blue, to sea-green, and then disappears.
500	932	Commences to be covered with a light coating of oxide; loses a good deal of its hardness, becomes much more impressible to the hammer, and can be twisted with ease.
525	977	Becomes nascent red.
700	1292	Sombre red.
800	1472	Nascent cherry.
900	1657	Cherry.
1000	1832	Bright cherry.
1100	2012	Dull orange.
1200	2192	Bright orange.
1300	2372	White.
1400	2552	Brilliant white—welding heat.
1500 1600	2732 2912	Dazzling white.

TABLE OF DECIMAL EQUIVALENTS OF ONE INCH.

1/64	.015625	17/64	.265625	33/64	.515625	49/64	.765625
1/32	.03125	9/32	.28125	17/32	.53125	25/32	.78125
3/64	.046875	19/64	.296875	35/64	.546875	51/64	.796875
1/16	.0625	**5/16**	.3125	**9/16**	.5625	**13/16**	.8125
5/64	.078125	21/64	.328125	37/64	.578125	53/64	.828125
3/32	.09375	11/32	.34375	19/32	.59375	27/32	.84375
7/64	.109375	23/64	.359375	39/64	.609375	55/64	.859375
1/8	.125	**3/8**	.375	**5/8**	.625	**7/8**	.875
9/64	.140625	25/64	.390625	41/64	.640625	57/64	.890625
5/32	.15625	13/32	.40625	21/32	.65625	29/32	.90625
11/64	.171875	27/64	.421875	43/64	.671875	59/64	.921875
3/16	.1875	**7/16**	.4375	**11/16**	.6875	**15/16**	.9375
13/64	.203125	29/64	.453125	45/64	.703125	61/64	.953125
7/32	.21875	15/32	.46875	23/32	.71875	31/32	.96875
15/64	.234375	31/64	.484375	47/64	.734375	63/64	.984375
1/4	.25	**1/2**	.50	**3/4**	.75	1	

MELTING-POINT OF METALS, ETC.

Names.	Fahr.	Names.	Fahr.
Platina	4590°	Wrought iron	2900°
Antimony	842	Steel	2500
Bismuth	487	Copper	2000
Tin	475	Glass	2377
Lead	620	Beeswax	151
Zinc	700	Sulphur	239
Cast iron	2100	Tallow	92

TABLE 5.

Weight of Various Substances.

Rule.—Divide the specific gravity of the substance by 16 and the quotient will give the weight of a cubic foot of it in pounds.

Substances—Metals.	Specific Gravity.	Weight of a Cubic Inch.	Substances—Metals.	Specific Gravity.	Weight of a Cubic Inch.
Aluminum	2,560	.0926	Lead, cast	11,352	.4106
Brass, plate	8,380	.3031	Lead, rolled	11,388	.4119
Brass, wire	8,214	.2972	Mercury, + 32°	13,598	.4918
Bronze, gun-metal	8,700	.3147	Mercury, 60°	13,580	.4912
Copper, cast	8,788	.3179	Mercury, 212°	13,370	.4836
Copper, plates	8,698	.3146	Steel, plates	7,806	.2823
Copper, wire	8,880	.3212	Steel, soft	7,833	.2833
Iron, cast	7,207	.2607	Steel, wire	7,847	.2838
Iron, cast, gun-metal	7,308	.264	Tin, Cornish, hammered	7,390	.2673
Iron, wrought bars	7,788	.2817	Zinc, cast	6,861	.2482
Iron, rolled plates	7,704	.2787	Zinc, rolled	7,191	.26

TABLE 6.

Weight of Timber per Cubic Foot.

Ash	46 lbs.	Mahogany, Honduras	35 lbs.
Beech	44 "	" Spanish	53 "
Birch	45 "	Oak, English	54 "
Boxwood	62 "	Pine, red	30 to 44 "
Elm	34 "	" yellow	29 to 41 "
Larch	34 "	" white	30 "
Lignum-vitæ	80 "	Teak	41 to 55 "

CIRCUMFERENCES AND AREAS OF CIRCLES ADVANCING BY EIGHTHS.

Diam.	Circum.	Area.
1/64	.04909	.00019
1/32	.09818	.00077
3/64	.14726	.00173
1/16	.19635	.00307
3/32	.29452	.00690
1/8	.39270	.01227
5/32	.49087	.01917
3/16	.58905	.02761
7/32	.68722	.03758
1/4	.78540	.04909
9/32	.88357	.06213
5/16	.98175	.07670
11/32	1.0799	.09281
3/8	1.1781	.11045
13/32	1.2763	.12962
7/16	1.3744	.15033
15/32	1.4726	.17257
1/2	1.5708	.19635
17/32	1.6690	.22166
9/16	1.7671	.24850
19/32	1.8653	.27688
5/8	1.9635	.30080
21/32	2.0617	.33824
11/16	2.1598	.37122
23/32	2.2580	.40574
3/4	2.3562	.44179
25/32	2.4544	.47937
13/16	2.5525	.51849
27/32	2.6507	.55914
7/8	2.7489	.60132
29/32	2.8471	.64504
15/16	2.9452	.69029
31/32	3.0434	.73708
1	3.1416	.7854
1/16	3.3379	.8866
1/8	3.5343	.9940
3/16	3.7306	1.1075
1/4	3.9270	1.2272
5/16	4.1233	1.3530
3/8	4.3197	1.4849
7/16	4.5160	1.6230
1/2	4.7124	1.7671
9/16	4.9087	1.9175
5/8	5.1051	2.0739
11/16	5.3014	2.2365
3/4	5.4978	2.4053
13/16	5.6941	2.5802
7/8	5.8905	2.7612
15/16	6.0868	2.9483
2	6.2832	3.1416
3/8	7.4613	4.4301
7/16	7.6576	4.6664
1/2	7.8540	4.9087
9/16	8.0503	5.1578
5/8	8.2467	5.4119

Diam.	Circum.	Area.
2 11/16	8.4430	5.6727
3/4	8.6394	5.9396
13/16	8.8357	6.2126
7/8	9.0321	6.4918
15/16	9.2284	6.7771
3	9.4248	7.0686
1/16	9.6211	7.3662
1/8	9.8175	7.6699
3/16	10.014	7.9798
1/4	10.210	8.2958
5/16	10.407	8.6179
3/8	10.603	8.9462
7/16	10 799	9.2806
1/2	10.996	9.6211
9/16	11.192	9.9678
5/8	11.388	10.321
11/16	11.585	10.680
3/4	11.781	11.045
13/16	11.977	11.416
7/8	12.174	11.793
15/16	12.370	12.177
4	12.566	12.566
1/16	12.763	12.962
1/8	12.959	13.364
3/16	13.155	13.772
1/4	13.352	14.186
5/16	13.548	14.607
3/8	13.744	15.033
7/16	13.941	15.466
1/2	14.137	15.904
9/16	14.334	16.349
5/8	14.530	16.800
11/16	14 726	17.257
3/4	14.923	17.728
13/16	15.119	18.190
7/8	15.315	18.665
15/16	15.512	19.147
5	15.708	19.635
1/16	15.904	20.129
1/8	16.101	20.629
3/16	16.297	21.135
1/4	16.493	21.648
5/16	16.690	22.166
3/8	16.886	22.691
7/16	17.082	23.221
1/2	17.279	23.758
9/16	17.475	24.301
5/8	17.671	24.850
11/16	17.868	25.406
6 1/8	19.242	29.465
1/4	19.635	30.680
3/8	20.028	31.919
1/2	20.420	33.183

Diam.	Circum.	Area.
6 5/8	20 813	34.472
3/4	21.206	35.785
7/8	21.598	37.122
7	21.991	38.485
1/8	22.384	39.871
1/4	22.776	41.282
3/8	23.169	42.718
1/2	23.562	44.179
5/8	23.955	45.664
3/4	24.347	47.173
7/8	24.740	48.707
8	25.133	50.265
1/8	25.525	51.849
1/4	25.918	53.456
3/8	26.311	55.088
1/2	26.704	56.745
5/8	27.096	58.426
3/4	27.489	60.132
7/8	27.882	61.862
9	28.274	63.617
1/8	28.667	65.397
1/4	29.060	67.201
3/8	29.452	69.029
1/2	29.845	70.882
5/8	30.238	72.760
3/4	30.631	74.662
7/8	31.023	76.589
10	31.416	78.540
1/8	31.809	80.516
1/4	32.201	82.516
3/8	32.594	84.541
1/2	32.987	86.590
5/8	33.379	88.664
3/4	33.772	90.763
7/8	34.165	92.886
11	34.558	95.033
1/8	34.950	97.205
1/4	35.343	99.402
3/8	35.736	101.62
1/2	36.128	103.87
5/8	36.521	106.14
3/4	36.914	108.43
7/8	37.306	110.75
12	37.699	113.10
1/8	38.092	115.47
1/4	38.485	117.86
3/8	38.877	120.28
1/2	39.270	122.72
5/8	39.663	125.19
3/4	40.055	127.68
7/8	40.448	130.19

To find the weight of castings by the weight of pine patterns, multiply the weight of the pattern by 12 for cast iron, 13 for brass, 19 for lead, 12.2 for tin, 14.4 for zinc, and the product is the weight of the casting.

CHAPTER I.*

SCREWS, NUTS, AND BOLTS.

A Screw is a helical projection or thread formed upon a cylinder and is the most common device used in mechanical

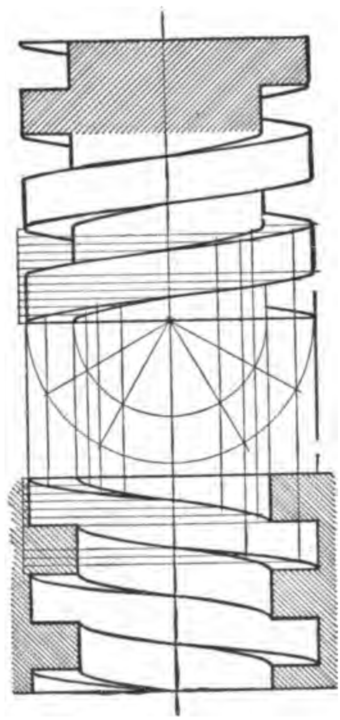

FIG. 28.

combinations. It is employed in the construction of machinery for producing pressure contact and transmitting

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motion. When the thread of an external screw is made to fit into the corresponding hollow of an internal screw (Fig. 28) the latter is termed its nut.

The Pitch of a Screw-thread is the lineal distance its nut would advance along the axis in one turn. In a single-threaded screw the pitch is the distance between the centres of two consecutive threads measured in the direction of the axis, in a double-threaded screw it is the distance from centre to centre of every alternate thread, and in a triple-threaded screw it is a distance that will embrace three threads. For screw-fastenings, instead of giving the pitch the number of threads per inch of screw is given—for example, a bolt of 1″ diameter has generally 8 threads per inch; this means that the bolt has a single thread wound around it 8 times for every inch of its length.

Right- and Left-handed Screws.—Screws are made right- and left-handed, of which the right-handed are the more common and are distinguished by their nuts advancing along the screws when turned in the direction in which the hands of a watch revolve. On a drawing the right-handed screws are distinguished by the threads inclining upwards towards the right hand when the screws are in a vertical position, as in Fig. 28. When a nut with a right-handed thread is shown in section the direction of the threads in the nut is the opposite to the threads on the screw.

The Nominal Diameter of a Screw is the diameter over the tops of the threads and is equal to the diameter of the cylinder upon which the thread is cut. It is the area of the nominal diameter that is considered when estimating the shearing strength.

The Effective Diameter is the diameter at the bottom of the thread and is equal to the diameter of the hole in the nut before its threads are cut. Unless when the bolts are subjected to a shearing stress, it is the area of the effective diameter that is considered in estimating their strength.

The Depth of the Thread is the distance measured perpendicularly to the axis of the screw from the top to the bottom of the thread.

NOTATION.

$d =$ nominal diameter of bolt;
$d_1 =$ effective diameter of bolt;
$\delta =$ depth of thread;
$\delta_1 =$ total depth of V;
$p =$ pitch of thread;
$n =$ number of threads per inch.

The Forms of Screw-threads in general use in machine construction are represented in Figs. 29–33. The V thread is adopted on all screw-fastenings because of the shearing strength of the threads and frictional holding power, which is due to the normal pressure on the thread being inclined

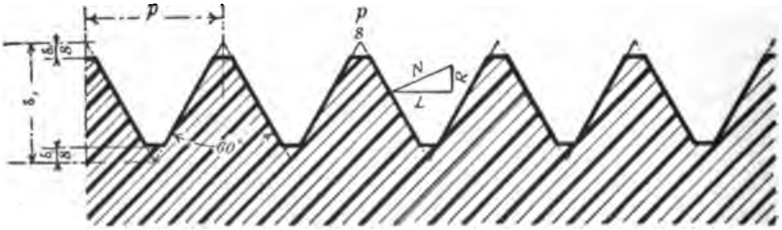


FIG. 29

to the axis of the screw. This normal force N, Fig. 29, may be resolved into two components, one L parallel to the

axis of the screw, and the other R at right angles to it. L represents the load carried by the thread and R the force tending to burst the nut; therefore the greater the angle of the V the greater will be the normal component or bursting force, and, the friction being proportional to the normal force, it will increase with the angle of the V. Of the forms of V threads shown two (Figs. 29 and 33) are in common use in the United States for bolts and nuts.

The Sellers or United States Standard, a section of which is shown in Fig. 29, has been adopted by the U. S. Government, the Railway Master Mechanics' Association, the Master Car-builders' Association, and many of the principal manufactories in this country. The sides of this thread form an angle of 60° with each other, and are $\frac{1}{8}$ of δ_1 short of meeting at a sharp point at the tops and bottoms, which makes the sides of the thread in length equal to $\frac{3}{4}$ of the pitch, and the depth of thread δ will be expressed by the formula

$$\delta = \tfrac{3}{4} \times p \sin 60° = 0.65p. \quad . \quad . \quad . \quad . \quad (1)$$

The effective diameter will then be

$$d_1 = d - 2\delta = d - 1.3p = d - \frac{1.3}{n}. \quad . \quad . \quad (2)$$

The relation between the pitch and the diameter will be expressed by the formula

$$p = 0.24\sqrt{d + 0.625} - 0.175. \quad . \quad . \quad . \quad (3)$$

The number of threads per inch is

$$n = \frac{1}{p} = \frac{1}{0.24\sqrt{d + 0.625} - 0.175}. \quad . \quad . \quad (4)$$

The table of proportions on page 70 has been deduced from the preceding formulæ. A difference, however, may be found between the formulæ and the table in the number of threads per inch, as the table has been modified to avoid as far as practicable troublesome combinations in the gears of screw-cutting machines.

Exercise 1.—Draw 6 threads in sectional outline, of the *Sellers thread* (Fig. 29), suitable for a screw 6″ in diameter. *Scale three times full size.*

Construction.—Begin by drawing a horizontal line in the upper left-hand corner of the paper $\frac{3}{4}''$ down from the border-line, and a vertical line about $\frac{3}{4}''$ in from the left-hand border-line. Then find the pitch p by the formula (3), and from where the two lines you have just drawn intersect mark off with the scale on the horizontal line 6 points a distance apart equal to the pitch as found by the formula. Through these points with the 30° triangle draw the V's. Complete the pencilling by dividing the depth of the V into 8 equal divisions, and cut off one division at the top and bottom of each thread.

The Sharp V Thread, shown in Fig. 30, is one of the

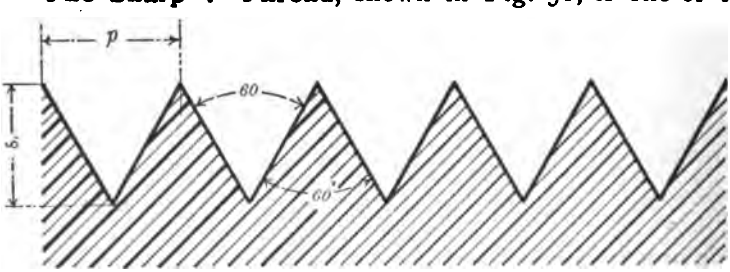


FIG. 30.

forms of threads that were in use before the Sellers thread

was adopted as the U. S. standard, and is still used, although condemned by all progressive engineers. This thread is the same as the Sellers thread except that the sides are made to meet at a sharp point at the top and bottom, which makes the sides of the thread equal in length to the pitch p, and the depth of the thread δ will be expressed by the formula.

$$\delta = p \sin 60° = 0.866p. \quad . \quad . \quad . \quad . \quad (5)$$

The effective diameter of the bolt (d_1) will then be expressed by the formula

$$d_1 = d - 2 \times 0.866p = d - 1.732. \quad . \quad . \quad (6)$$

Now, comparing the effective diameters, we have:

U. S. threads $\qquad d_1 = d - 1.3p. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$

V threads $\qquad d_1 = d - 1.732p. \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$

This serves to show that with an equal pitch the *effective diameter* of the screw having a U. S. standard thread is greater than one with a sharp V thread. While the latter form of thread materially diminishes the strength of the bolt, the sharp point adds very little strength to the thread. A further objection to this form of thread is the variation in depth of the threads due to the wear of the sharp points on the taps and dies used in producing them.

The Whitworth V Thread, an outline section of which is shown in Fig. 31, is the British standard, and is generally adopted on all screw-fastenings in British machine construction. It has the sides of the V inclined to each other at an angle of 55°, and has an amount rounded off at the top and bottom equal to $\frac{1}{6}$ of the total depth of the V. *The table of*

dimensions for Whitworth screws (page 70) has been deduced from the following formulæ. The total depth of the V

$$\delta_1 = 0.5 \cot 27\tfrac{1}{2}^\circ = 0.96p. \quad . \quad . \quad . \quad . \quad . \quad (7)$$

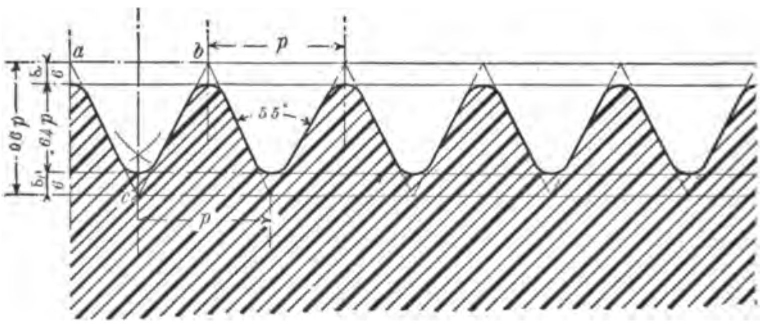


Fig. 31.

The depth of the finished thread

$$\delta = \tfrac{2}{3} \times 0.96p = 0.64p. \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The pitch $\quad p = 0.08d + 0.04. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$

Number of threads per inch

$$= \frac{1}{p} \quad \text{and} \quad p = \frac{1}{n}. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The diameter at the bottom of the thread will be given by the formula

$$d_1 = d - 2 \times 0.64p = d - \frac{1.28}{n}. \quad . \quad . \quad . \quad . \quad (11)$$

Exercise 2.—Draw 6 threads of the *Whitworth form of thread* (Fig. 31). Pitch ½″. *Scale three times full size.*

Construction.—At a suitable distance below the drawing of the Sellers thread draw two horizontal lines parallel to each other and a distance apart equal to 0.96*p*. On the upper line mark off a distance *ab* equal to the pitch. Bisect

ab and draw the bisecting line to cut the lower parallel line at the point *c*. Join *ca* and *cb*, which will be inclined to each other at an angle of 55°. Mark off the pitch from *b* along the upper line, and from *c* along the lower line, to give the required number of threads. Complete the pencilling by rounding off the sharp points of the V.

The Square Screw-thread.—The form of thread which is invariably called the square thread is really a rectangle, the depth of the thread being equal to $0.485p$ and its width equal to $0.5p$. However, it is usual and accurate enough to make it square upon the drawing. On screws of the same diameter the pitch of a square-threaded screw is usually made equal to twice the pitch of one with a V thread; therefore the square thread will have only half the amount of material at the bottom of the thread that the V thread has to resist the shearing action of the load. As the bearing-surfaces of this screw are perpendicular to the axis, and the force applied parallel to it, there will be no bursting force upon the nut; and as the reaction is nearly equal to the load on the square-threaded screw, there will be less friction than there is under the same conditions with a V thread; consequently the square thread is best adapted for transmitting motion when the load has to be moved in opposite directions.

The Knuckle or Rounded Screw-thread is a modification of the square thread in which the top and bottom of each thread are made semicircular, as shown in Fig. 32. This form of thread is used for rough work and can be readily thrown in and out of gear with a portion of a nut.

The Buttress Screw-thread is a combination of the V and square threads, one side being perpendicular, and the

other inclined at an angle of 45° to the axis of the screw, and has an amount cut from the top and bottom of each

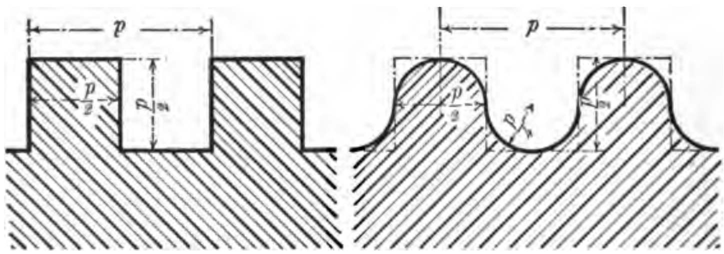


FIG. 32.

thread equal to $\frac{1}{8}$ of the total depth of the thread, as shown in Fig. 33. This form of thread can be used only when the pressure is on that side of the thread which is at right angles to the axis of the screw.

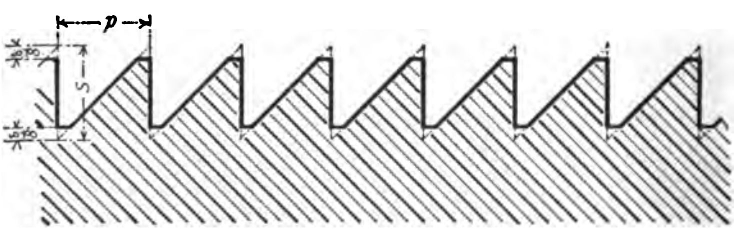


FIG. 33.

Exercise 3.—Draw the sectional outline of the square, knuckle, and buttress threads shown in Figs. 32 and 33. Pitch 1″. *Scale twice full size.*

Pipe-threads.—Previous to the year 1862 no common system had been agreed upon for the form or proportions of pipe-threads. Since that time, owing to the efforts of the late Robert Briggs, C.E., who proposed formulæ and tables for the dimensions of pipes and pipe-threads, a standard

TABLE 1.

STANDARD DIMENSIONS OF WROUGHT-IRON WELDED TUBES.

(BRIGGS STANDARD.)

Diameter of Tube.			Thickness of Metal.	Screwed Ends.	
Nominal Inside.	Actual Inside.	Actual Outside.		Number of Threads per Inch.	Length of Perfect Screw.
Inches.	Inches.	Inches.	Inch.	No.	Inches.
⅛	0.270	0.405	0.068	27	0.19
¼	0.364	0.540	0.088	18	0.29
⅜	0.494	0.675	0.091	18	0.30
½	0.623	0.840	0.109	14	0.39
¾	0.824	1.050	0.113	14	0.40
1	1.048	1.315	0.134	11½	0.51
1¼	1.380	1.660	0.140	11	0.54
1½	1.610	1.900	0.145	11½	0.55
2	2.067	2.375	0.154	11½	0.58
2½	2.468	2.875	0.204	8	0.89
3	3.067	3.500	0.217	8	0.95
3½	3.548	4.000	0.226	8	1.00
4	4.026	4.500	0.237	8	1.05
4½	4.508	5.000	0.246	8	1.10
5	5.045	5.563	0.259	8	1.16
6	6.065	6.625	0.280	8	1.26
7	7.023	7.625	0.301	8	1.36
8	7.982	8.625	0.322	8	1.46
9	9.000	9.625	0.344	8	1.57
10	10.019	10.750	0.366	8	1.68

Taper of conical tube-ends, 1 in 32 to axis of tube (¾ in. per foot total taper).

system has been generally used and was formally adopted by the manufacturers of wrought-iron pipes and boiler-tubes and by the Association of Manufacturers of Brass and Iron Steam-, Gas-, and Water-work of the United States.

The following is an extract from a paper by Mr. Briggs as given in the report of the American Society of Engineers:

"The thread employed has an angle of 60°; it is slightly rounded off, both at the top and at the bottom, so that the height or depth of the thread, instead of being exactly equal to the pitch, is only four fifths of the pitch, or equal to $0.8\frac{1}{n}$,

if n be the number of threads per inch. For the length of tube-end throughout which the screw-thread continues perfect the empirical formula used is $T = (0.8D + 4.8) \times \frac{1}{n}$, where D is the actual external diameter of the tube throughout its parallel length, and is expressed in inches. Further back, beyond the perfect threads, come two having the same taper at the bottom, but imperfect at the top. The remaining imperfect portion of the screw-thread, furthest back from the extremity of the tube, is not essential in any way to this system of joint; and its imperfection is simply incidental to the process of cutting the thread at a single operation.

Exercise 4.—Draw a section of a pipe-screw (Fig. 34) for a wrought-iron pipe 8″ in diameter. *Scale five times full size.*

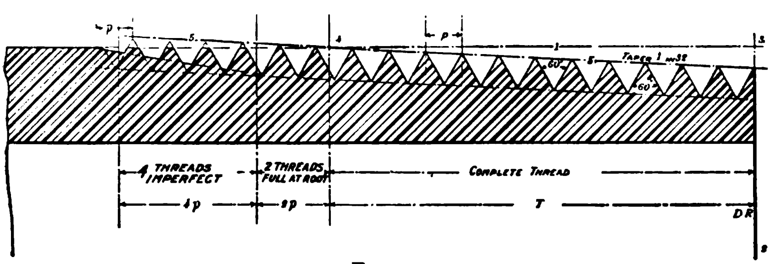


FIG. 34.

Construction.—Draw two lines parallel to each other at a distance apart equal to the thickness of metal as given in the table; then draw the vertical line 2 to represent the end of the pipe, and from 2 along the line 1 mark off 3, 4, equal to T. *Taper 1 in 32 means an inclination of 1 unit in height to every 32 units in length.* From the point 4 draw the line 5 at the required inclination. On the line 5 from where it intersects 2 mark off points at a distance apart equal to the pitch, and through these points with the 30° triangle draw the

threads. The bottoms of the last 4 threads are cut off by drawing a line from the bottom of the last thread that is full at the bottom to a point on the surface of the pipe which is a distance beyond the screwed part equal to the pitch.

Screw-thread Conventions.—The method of drawing screws to represent their true form is shown in Fig. 28, but it is quite obvious that it is unnecessary for the draftsman to perform this lengthy geometrical construction to indicate each screwed piece upon the drawing. Instead he adopts some convention suitable to the class of drawing he is making that can be quickly drawn and is generally understood to represent a screw-thread. Fig. 35, No. 1,

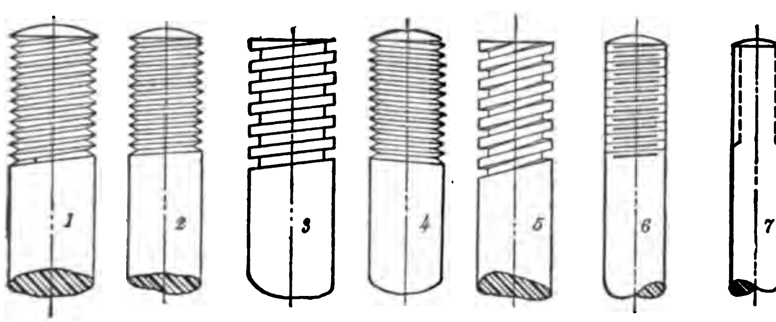


FIG. 35.

shows a convention for a double V thread; No. 2, a single V thread; No. 3, a single square thread; No. 4, a single left-hand V thread; No. 5, a double right-hand square thread; No. 6, any V thread of small diameter; No. 7, any thread of very small diameter. The method adopted on rough drawings and sketches is shown at No. 7. The dotted lines indicate the bottom of the thread, and the distance they extend along the piece the length of the

screwed part. At Nos. 1, 2, 4 are shown conventions adopted upon finished drawings to represent threaded screws of a large diameter and wide pitch. There are various ways of improving the appearance of this convention: one is by shading the lower lines of each thread, as shown in Fig. 37, and another method is to fill in completely the under side of the thread, as shown in Fig. 39. At No. 6 is shown a method adopted on working drawings to represent screw-threads upon pieces of a small diameter or large screws drawn to a small scale. Here the narrow lines indicate the top and the wide lines the bottom of the screw-thread. When a very long screw has to be represented upon a drawing, as is often the case with the square-threaded screw, a few threads are drawn at the beginning of the screwed part, and the length of the screw is indicated by dotted lines drawn from the bottoms of the threads.

The Nut.—The most common application of the screw for producing contact pressure is the bolt, used in conjunction with a nut, of which there are different forms. The form most in use is the hexagonal (Fig. 37).

The standard proportions for hexagonal nuts are:

H = height = diameter of bolt (d).

F = distance across the flats = $1\frac{1}{2}d + \frac{1}{8}$ of an inch.

D = distance across the corners = $(1\frac{1}{2}d + \frac{1}{8}'')\ 1.155$.

Fig. 35 shows the true form of the curves when the end of the nut is machined to form a part of a sphere or cone. This rounding or bevelling off of the corners is called chamfering. The radius r of the chamfering is made from $1\frac{1}{2}d$ to $2d$, and the angle a is made from 60° to 45° with the axis of the nut. *When representing nuts upon a drawing they should*

always be drawn to show the distance across the angles, as in Fig. 40.

Exercise 5.—Draw the true curves of a hexagonal nut for a bolt 6″ in diameter when the top of the nut is chamfered

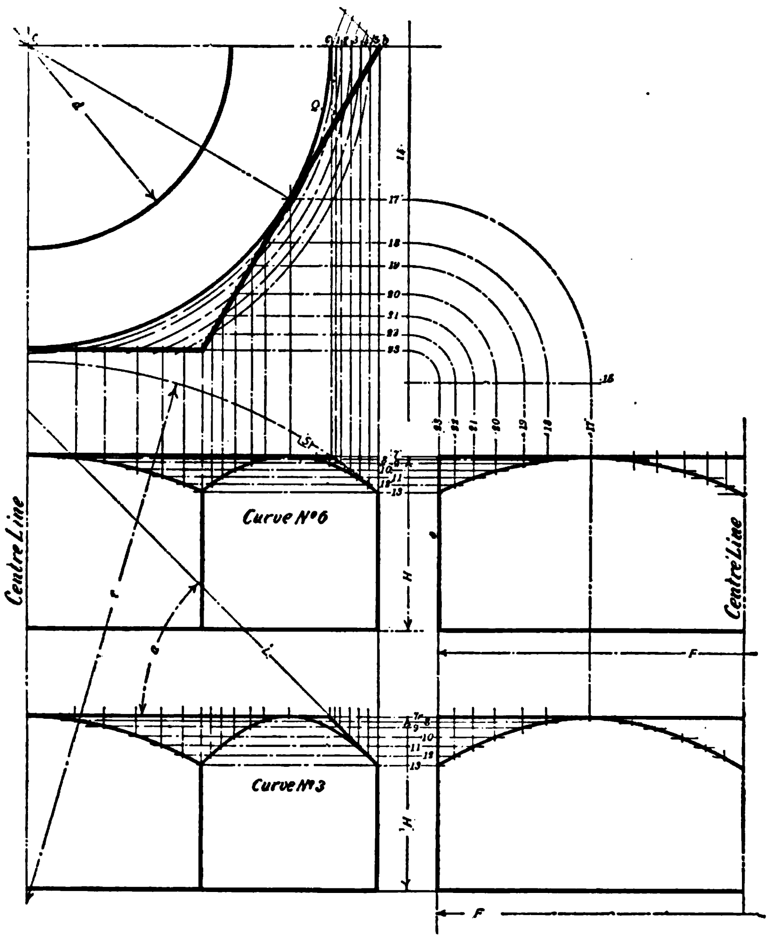


FIG. 36.

off to form a part of a sphere with a radius $r = 1\frac{1}{2}$ times the diameter of the bolt (d), and when the chamfering is a part

of a cone the side of which makes an angle of 45° with the axis of the nut, as shown in Fig. 36.

Construction.—Begin with the plan, first locating the centre *c*, and with *c* as a centre and a radius equal to $\frac{1}{2}d$ draw the quadrant representing the hole in the nut, and from the same centre and a radius equal to half the distance across the flats *F* draw the quadrant *Q*, and on this quadrant circumscribe a part of a hexagon with the 30° triangle and T square, as shown in Fig. 37. Draw the part elevations and end views, and with *r* as a radius and the centre on the centre line draw the arc *S*, which represents the spherical chamfer, and on the lower elevation draw the angle *a*. Divide *eb* into any number of divisions, say 6, at points 1, 2, 3, 4, 5*d*. Where perpendicular lines drawn through these points intersect the arc *S* and line *L* draw the horizontal lines 7, 8, 9, 10, 11, 12, 13, and with *c* as a centre and radii *c*1, *c*2, *c*3, *c*4, *c*5 draw arcs, and from where these arcs intersect the inclined face of the nut draw vertical lines to intersect the lines 7, 8, 9, 10, etc. These points of intersection will be points of the curve on the side face of the nut. The curve of the front face will be an arc of a circle. To find the curves on the side view draw a line 15 say $\frac{1}{2}''$ below and parallel to the lower face of the nut in plan, and a perpendicular line 14 half an inch to the left of the end view; where the arcs drawn through the points 1, 2, 3, etc., from the centre *c* cut the inclined face of the nut in plan draw horizontal lines to intersect the line 14; and with a centre at the intersection of the lines 14 and 15 revolve the lines 17, 18, 19, 20, 21, 22, 23 on to the line 15 and draw perpendicular lines through the points of intersection. The line 17 revolved will be the cen-

tre of the nut face on the end view, and the intersection of the lines 17, 18, 19, 20, 21, 22, 23 with the horizontal lines 7, 8, 9, 10, 11, 12, 13 will be points on one half of the required curve. To complete the curve, with a centre at the intersection of the line 17 and the top of the nut mark with the compasses corresponding points on the other side of the line 17.

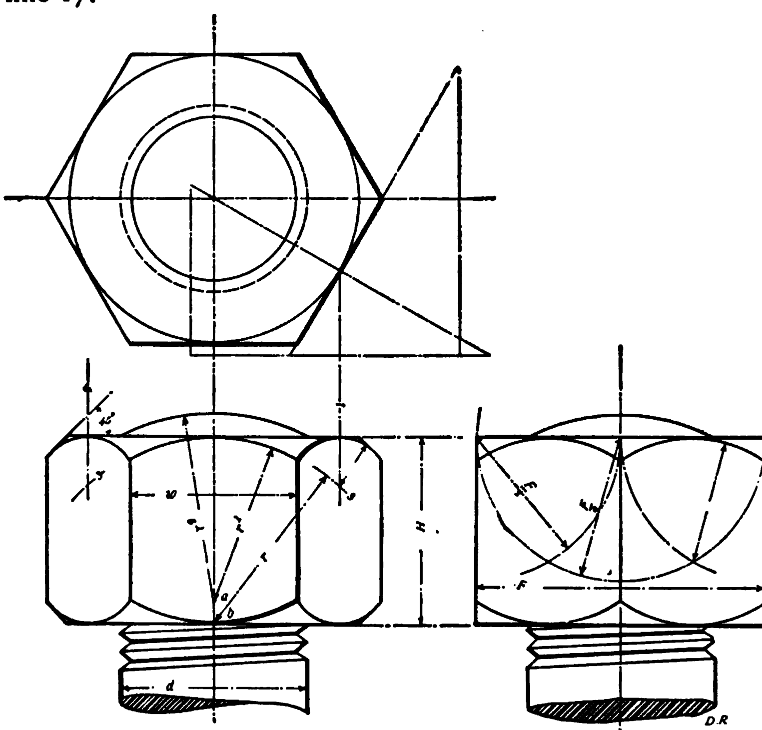


FIG. 37.

A Conventional Method of representing large nuts on drawings is shown in Fig. 37. In this representation the curves of the nut are arcs of circles and the corners are chamfered off at an angle of 45° to the axis of the nut.

Exercise 6.—Draw the three views for a bolt $1\frac{1}{8}''$ in diameter. *Scale full size.*

Construction.—Begin, as in the last exercise, by drawing the plan. Locate the centre and draw a circle equal in diameter to the distance across the flats $1\frac{1}{2}d + \frac{1}{8}''$; on this circle with the set-square circumscribe a hexagon, and find the centre of the side faces in the manner shown. Draw the elevation and end view of the hexagon without the curves. With centre a and radius r' equal to w draw the arc of the middle face tangent to the top of the nut, and with centre b and radius r equal to d draw the arcs 3 to intersect the lines 1 and 2. These points of intersection will be the centres of the arcs of the side face. The method of finding the centres of the curves on the end view is clearly shown on the drawing. Through the points where the outside diameter of the bolt intersects the top of the nut with a radius $r^2 = d$ draw the arc representing the bolt-point.

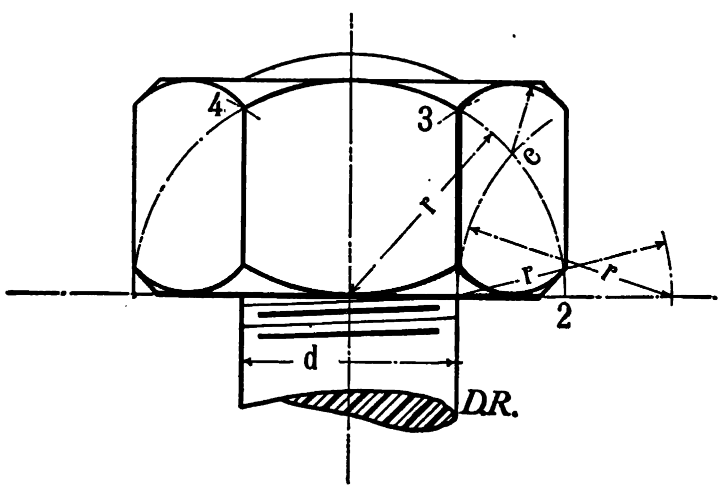


FIG. 38.

When representing small nuts or nuts drawn to a small scale, it is usual to make the distance across the angles $= 2d$. This method does not give the correct proportions and should only be used on nuts and bolt-heads when d is less than 1″ in diameter when drawn to scale. When nuts are chamfered on the upper side only, it is usual to cut the corners parallel to the axis, thus leaving a cylindrical projection on the under side, which bears on the piece the nut is holding, as shown in Fig. 39.

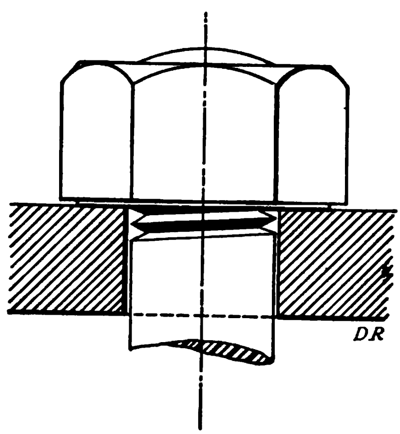


Fig. 39.

The diameter of the cylindrical projection is equal to the distance across the flats (i.e., $1\frac{1}{2}d + \frac{1}{8}''$).

Exercise 7.—Draw a hexagonal nut for a bolt $\frac{3}{4}''$ in diameter chamfered on the upper side and finished on the under side as shown in Fig. 39. Make the distance across the angles $= 2d$, and draw the curves by the method shown in Fig. 38. *Scale full size.*

Construction.—Draw the semicircle 1, 2 and divide it into

three equal divisions at the points 1, 2, 3, 4; through these points draw perpendicular lines to intersect the top of the nut. The method of finding the centres of the arcs of the side faces will be clearly understood from Fig. 38, where r *is* in each case $= d$.

Machine fastenings are most commonly effected by means of bolts, keys, or rivets. When two or more pieces have to be held together with the intention of disconnecting them again, a bolt or key is used; the rivet being used only when the connection is to be permanent. The most common form

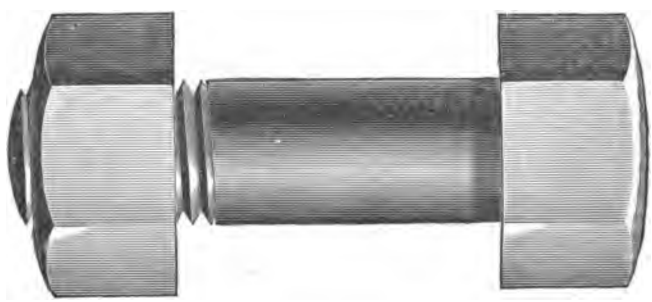


FIG. 40.

of bolt used in general machine construction is the *hexagonal, headed bolt* shown in Fig. 40.

Exercise 8.—Draw a hexagonal headed bolt and nut in position on a cast-iron pipe-flange (Fig. 41). Make the bolt $\frac{3}{4}''$ in diameter. *Scale full size.*

Construction. — First draw the lines representing the thickness of the pipe and flanges. The angles of the nut should be clear of the fillet about $\frac{1}{8}''$ and the radius (r) should be at least $\frac{1}{8}''$. Therefore, the distance (b) will be equal to half the distance across the angles $+ \frac{1}{8}'' + r$. To give the flange a proper finish, the distance (a) is made from $\frac{1}{8}''$ to $\frac{1}{4}''$

greater than the distance across the angles of the nut. The number of threads per inch will be found in Table 8.

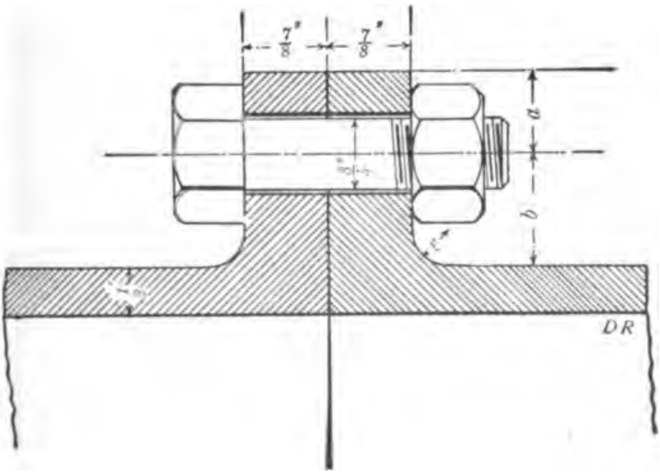


FIG. 41.

The Square-headed Bolt.—Fig. 42 is a cheaper make than the hexagonal and is generally used in structures of rough iron. It is sometimes, however, adopted in machine- and engine-construction generally when the head is let into a recess, as shown in Fig. 42. It is used in this instance in preference to the hexagonal head, because it is easier to make the square recess in the pattern. In Fig. 42, it is shown in combination with a square nut, the sides of which give a better gripping surface for the wrench than the hexagonal, but the latter can be screwed up in a more confined position, as it is only necessary to turn it through an angle of 60° to get the wrench or spanner on to the next two parallel faces; while the square nut has to be turned through an angle of 90° under the same conditions.

Exercise 9.—Draw a bolt with a square head, and nut, as shown in Fig. 42. Make the bolt 1″ in diameter. *Scale full size.*

Construction.—The proportions of heads and nuts will be found in Table 8. The radius (r) is made equal to F and tangent to the top of the nut or head.

A Stud-bolt consists of a bar screwed at both ends (Fig. 43), one end being screwed into the piece upon which the connection is made. The other piece is then passed over the studs and secured by a nut. To allow the nut to make a tight joint, the length of the body or plain part must always be less than the thickness of the piece into which it passes.

Studs are used only when it is impossible, or at least very inconvenient, to use an ordinary bolt. When studs are screwed into cast material, the screwed part should extend into the metal at least 1½ times their diameter, *and should never be allowed to bear on the bottoms of the holes.*

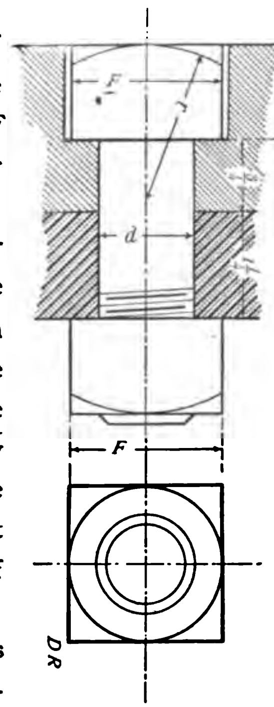


FIG. 42.

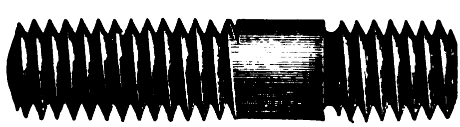


FIG. 43.

Fig. 44 shows a stud used to secure the cylinder-cover (c) to

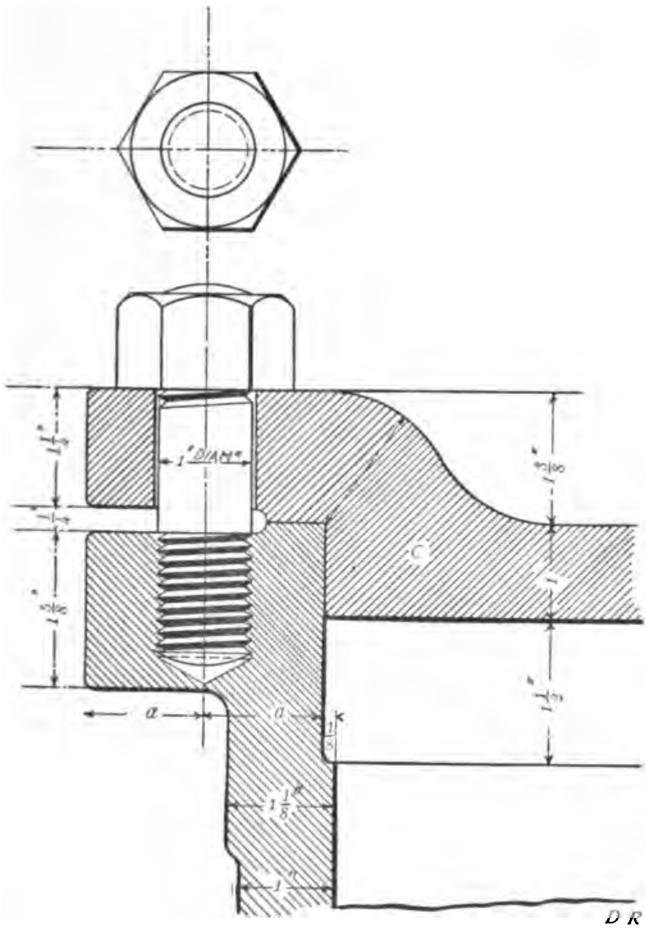


FIG. 44.

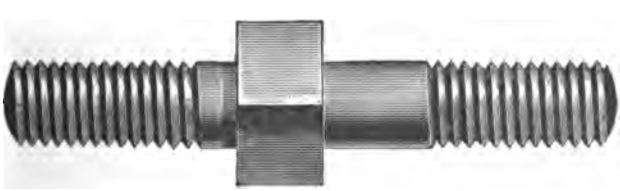


FIG. 45.

TABLE

UNITED STATES STANDARD OF

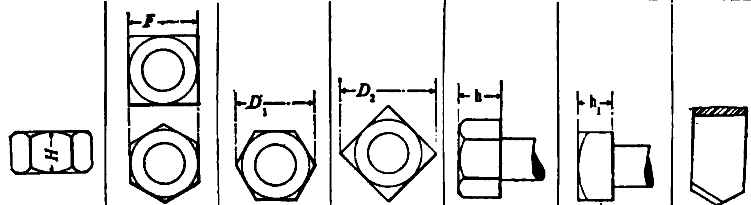
Screw-threads.

Diameter of Screw.	Number of Threads per Inch.	Diameter at Bottom of Threads.	Area at Bottom of Threads in Square Inches.	Area of Bolt Body in Square Inches.
¼	20	.185	.027	.049
5/16	18	.240	.045	.077
⅜	16	.294	.068	.110
7/16	14	.344	.093	.150
½	13	.400	.126	.196
9/16	12	.454	.162	.249
⅝	11	.507	.202	.307
¾	10	.620	.302	.442
⅞	9	.731	.420	.601
1	8	.837	.550	.785
1⅛	7	.940	.694	.994
1¼	7	1.065	.893	1.227
1⅜	6	1.160	1.057	1.485
1½	6	1.284	1.295	1.767
1⅝	5½	1.389	1.515	2.074
1¾	5	1.491	1.746	2.405
1⅞	5	1.616	2.051	2.761
2	4½	1.712	2.302	3.142
2¼	4½	1.962	3.023	3.976
2½	4	2.176	3.719	4.909
2¾	4	2.426	4.620	5.940
3	3½	2.629	5.428	7.069
3¼	3½	2.879	6.510	8.296
3½	3¼	3.100	7.548	9.621
3¾	3	3.317	8.641	11.045
4	3	3.567	9.963	12.566
4¼	2⅞	3.798	11.329	14.186
4½	2¾	4.028	12.753	15.904
4¾	2⅝	4.256	14.226	17.721
5	2½	4.480	15.763	19.635
5¼	2½	4.730	17.572	21.648
5½	2⅜	4.953	19.267	23.758
5¾	2⅜	5.203	21.262	25.967
6	2¼	5.423	23.098	28.274

NOTE.—The above table gives the sizes of the rough nuts and bolt-heads. The finished

8.

SCREW-THREADS, BOLTS, AND NUTS.



Nuts.				Heads.		Tap Drill.
1/4	1/2	37/64	7/10	1/4	1/4	3/16
5/16	19/32	11/16	10/12	5/16	19/64	1/4
3/8	11/16	51/64	63/64	3/8	11/32	5/16
7/16	25/32	9/10	1 7/64	7/16	25/64	23/64
1/2	7/8	1	1 15/64	1/2	7/16	13/32
9/16	31/32	1 1/8	1 23/64	9/16	31/64	15/32
5/8	1 1/16	1 7/32	1 1/2	5/8	17/32	17/32
3/4	1 1/4	1 7/16	1 49/64	3/4	5/8	5/8
7/8	1 7/16	1 21/32	2 1/32	7/8	23/32	3/4
1	1 5/8	1 7/8	2 19/64	1	13/16	27/32
1 1/8	1 13/16	2 3/32	2 9/16	1 1/8	29/32	31/32
1 1/4	2	2 5/16	2 53/64	1 1/4	1	1 3/32
1 3/8	2 3/16	2 17/32	3 3/32	1 3/8	1 3/32	1 3/16
1 1/2	2 3/8	2 3/4	3 23/64	1 1/2	1 3/16	1 9/32
1 5/8	2 9/16	2 31/32	3 5/8	1 5/8	1 9/32	1 13/32
1 3/4	2 3/4	3 3/16	3 57/64	1 3/4	1 3/8	1 1/2
1 7/8	2 15/16	3 13/32	4 5/32	1 7/8	1 15/32	1 5/8
2	3 1/8	3 5/8	4 27/64	2	1 9/16	1 3/4
2 1/4	3 1/2	4 1/16	4 61/64	2 1/4	1 3/4	1 31/32
2 1/2	3 7/8	4 1/2	5 31/64	2 1/2	1 15/16	2 3/16
2 3/4	4 1/4	4 29/32	6	2 3/4	2 1/8	2 7/16
3	4 5/8	5 3/8	6 17/32	3	2 5/16	2 5/8
3 1/4	5	5 13/16	7 1/16	3 1/4	2 1/2	2 29/32
3 1/2	5 3/8	6 7/32	7 39/64	3 1/2	2 11/16	3 1/32
3 3/4	5 3/4	6 21/32	8 1/8	3 3/4	2 7/8	3 11/32
4	6 1/8	7 3/32	8 41/64	4	3 1/16	3 19/32
4 1/4	6 1/2	7 9/16	9 3/16	4 1/4	3 1/4	3 13/16
4 1/2	6 7/8	7 31/32	9 3/4	4 1/2	3 7/16	4 1/32
4 3/4	7 1/4	8 13/32	10 1/4	4 3/4	3 5/8	4 9/32
5	7 5/8	8 27/32	10 49/64	5	3 13/16	4 1/2
5 1/4	8	9 9/32	11 21/64	5 1/4	4	4 3/4
5 1/2	8 3/8	9 23/32	11 7/8	5 1/2	4 3/16	4 31/32
5 3/4	8 3/4	10 5/32	12 3/8	5 3/4	4 3/8	5 7/32
6	9 1/8	10 19/32	12 15/16	6	4 9/16	5 7/16

sizes are: $H = d - 1/16''$; $F = 1\frac{1}{2}d + 1/16''$; $h = d - 1/16''$; $h_1 = \frac{1\frac{1}{2}d + 1/16''}{2}$.

the cylinder. Studs are preferred to bolts for this purpose because the flanges can be made very much smaller, and the cover can be removed and replaced without disturbing the cylinder-lagging. A stud should not be placed nearer to the edge of the metal than a distance equal to (*d*) measured from the centre of the stud, and in steam-tight joints it is usual to make the distance (*a*) equal to $1\frac{1}{4}d$, as shown in Fig. 44.

Fig. 43 shows the form of stud in general use. The body of this stud is made cylindrical and equal in diameter to the diameter of the screw. As the weakest part of the stud is at the change of section, the form of stud shown in Fig. 44, if subjected to a greater stress than it could withstand, would break off, leaving the screwed part in the metal, but by cutting a semicircular groove of a depth = the depth of the thread on the end of the body that comes in contact with the piece into which the stud is screwed, as in Fig. 43, this part is strengthened and the stud would then break where the upper screwed part joins the body. The broken stud can then be easily removed by means of a pipe-wrench. In Fig. 45, the stud has a square body which serves as a shoulder, against which the stud may be screwed up tight by means of a wrench applied to the square part. Studs with round bodies are screwed into position by means of a tool called a stud-nut; this consists of a long nut fitted with an internal screw, as shown in Fig. 46. To avoid damaging the point

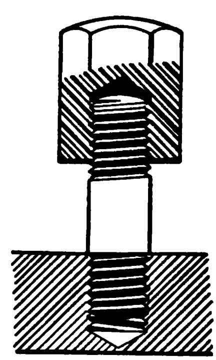


FIG. 46.

of the stud, the bottom of the screw in the stud-nut is lined with copper. By applying a wrench to the stud-nut, the stud can be screwed into the tapped hole in the metal until stopped by the plain portion on the stud. The stud-nut can then be removed by a quick turn back.

Exercise 10.—Draw a section of a steam-cylinder end-flange, showing the method of securing the cylinder-cover or head (c) to the cylinder (Fig. 44). *Scale full size.*

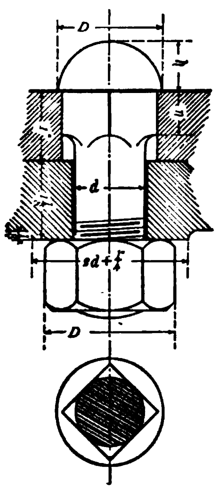


FIG. 47.

When a bolt-head is of such form, or in a position in which it cannot be held with a wrench to keep it from revolving when screwing up the nut, the bolt is provided with some device in the body to overcome the difficulty. The spherical or button-headed bolt, shown in Fig. 47, is provided with a square part under the head, which fits into a corresponding hole in the material through which it passes. Another design used for the same purpose is shown in Fig. 48; this is called a snug, and consists of one or two projections forged on the neck of the bolt and made to fit a correspondingly shaped hole in the metal.

Fig. 49 shows a bolt with a countersunk head and nut. The bolt is kept from revolving by a pin (p), which is driven into a hole drilled in the body of the bolt close to the head. The projecting part of the pin fits into a recess cut to receive it. The nut is provided with holes to receive the spanner used in screwing it up, and may be made equal in diameter

to half the amount of metal between the bottoms of the threads and the outside of the nut. The depth of the holes may be made .25 of *H*, the height of the nut. The project-

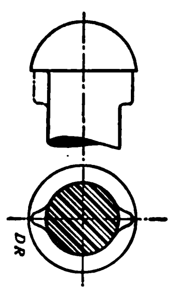


Fig. 48.

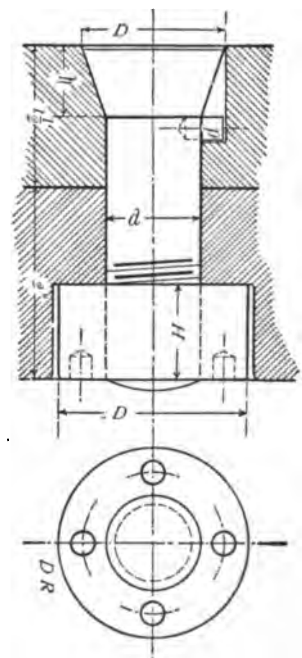


Fig. 49.

ing part of the pin (*p*) is usually made square and equal to .25*d*. The pin (*p*) is sometimes screwed into the bolt to avoid its being lost when the bolt is withdrawn.

The T-headed Bolt shown in Fig. 50 has the sides of the head level with the square neck or body of the bolt, and is used where there is not sufficient room to use bolts of the hexagonal or square-headed form. A common application of this form of bolt is shown in Fig. 50.

The **Tap-bolt** shown in Fig. 51 makes a fastening without the use of a nut. The bolt is screwed into a tapped hole in one of the pieces to be connected, while the head

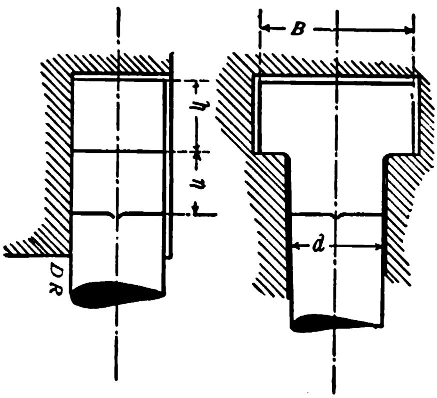


FIG. 50.

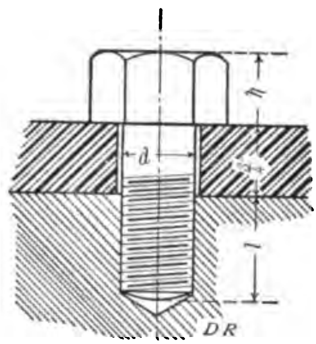


FIG. 51.

presses on the other piece. This form of bolt is used in place of a stud where the piece to be connected could not, if studs were used, be passed over the projecting studs, as in a pipe-fastening where two of the faces are at an angle to each other.

There is no standard for the foregoing bolt-heads and nuts, but the following proportions are in general use:

$$D = 1.5d, \quad D_1 = (1\tfrac{1}{2} + \tfrac{1}{8}'')1.55, \quad B = \text{from } 1\tfrac{1}{2}d + \tfrac{1}{8}'' \text{ to } 2d,$$

$$h = .7d, \quad H = d, \quad n = .6d, \quad l = 1\tfrac{1}{2}d.$$

Exercise 11.—Draw a *spherical* or *button-headed bolt* with a square neck; and a head with a snug on the neck, as shown in Figs. 47 and 48. A *countersunk-headed bolt* with a *countersunk nut* as shown in Fig. 49, a *T-headed bolt* with a square

neck, as shown in Fig. 50, and a *tap-bolt*, as shown in Fig. 51. Make *d* in each case = 1″. *Scale full size.*

Hook-bolt.—This form of bolt is used where it is impossible or undesirable to have bolt-holes through one of the connected pieces. A common application of this bolt is fastening pieces (such as hangers) to flanged beams, as shown in Fig. 52. To keep the bolt from turning, the body is

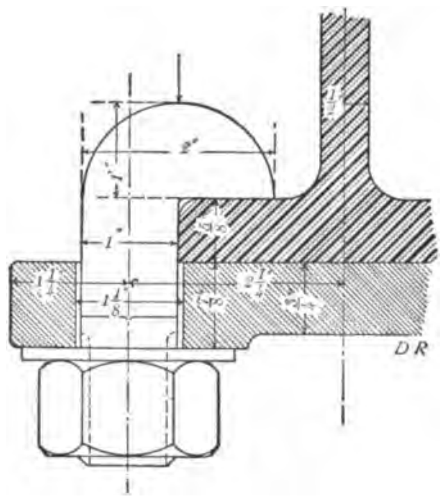


FIG. 52.

made square in cross-section and passes into a correspondingly shaped hole in the connected piece. The diameter of the screw is equal to the square body.

Exercise 12.—Draw an ELEVATION of a hook-bolt, fastening a piece to a flanged beam, as shown in Fig. 52, and PLAN of the bolt only, looking down on the bolt head. *Scale full size.*

Tapered Bolts are used to facilitate fitting where it is necessary that the bolt should be a perfect fit in the hole.

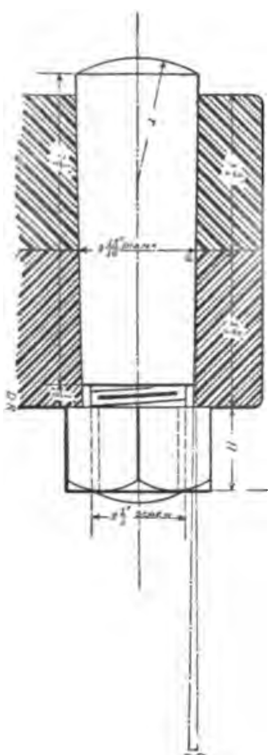


FIG. 53.

Fig. 53 shows a tapered bolt that is in common use in the couplings of propeller shafts of steamships. As coupling-bolts have only to resist the shearing force, caused by the twisting strain on the shaft, the diameter of the bolt is the diameter on the line where the two flanges come together, and its strength is equal to $\frac{\pi d^2}{4} f_s$.

As the screwed part of the bolt has only to resist the tension due to screwing up, this part is made smaller in diameter than the small end of the tapered part. In practice, the diameter of the screwed part is generally made equal to $\frac{7d + 1}{9}$, and the height of the nut from $\frac{1}{4}''$ to $\frac{1}{2}''$ less than the diameter of the screw. The advantages gained by using tapered instead of parallel bolts for couplings are: they can be made a perfect fit in the hole, which insures that the different lengths of shaft are in better alignment, are easier withdrawn, and, owing to the diameter of the screw being much smaller than the diameter at the junction of the shafts (i.e., the effective diameter), the flanges can be made smaller.

Exercise 13.—Draw a tapered bolt for a marine shaft-

coupling, showing a part of the shaft-flanges, to the dimensions given in Fig. 53. *Scale half size.*

Construction.—Draw the centre line of the bolt, then the line showing the junction of the flanges, and on this line mark off the diameter of the bolt. From the point (a) draw the line ab 12 inches long and parallel to the axis of the bolt, and from b draw bc perpendicular to ab and $\frac{3}{16}''$ long, join ac which makes the required taper. The radius (r) is equal to the diameter of the bolt at the large end.

Exercise 14.—Draw a tapered bolt as in the preceding exercise, leaving off the parts of the shafts, and making the diameter of the bolt 3 inches, and the length of the body equal to 8 inches. *Scale half size.*

Foundation-bolts.—This class of bolts is employed for fastening engine- and machine-frames to stone, brick, or concrete foundations.

The Rag-bolt (Fig. 54).—This form of bolt is fastened to stone by cutting a Lewis hole, which increases in size as it descends. The small end of the hole is made from $\frac{1}{4}''$ to $\frac{1}{2}''$ larger than the large end of the bolt-head. After the bolt-head is placed in the hole, the space between it and the sides of the hole is filled with molten lead or sulphur, thus securing the bolt firmly in position. The frame is cast with a projecting foot through which a hole is cored. This foot passes over the foundation-bolt and the engine- or machine-frame is held in position by the pressure of the nut. The diameter of the hole through the foot is $= d + \frac{1}{4}''$. The diameter of the washer w is equal to $2d + \frac{1}{2}''$, and the thickness $\frac{3}{20}$ of d. The distance a is $=$ half the diameter of the washer $+ \frac{3}{8}''$. The section of the bolt-head is oblong and

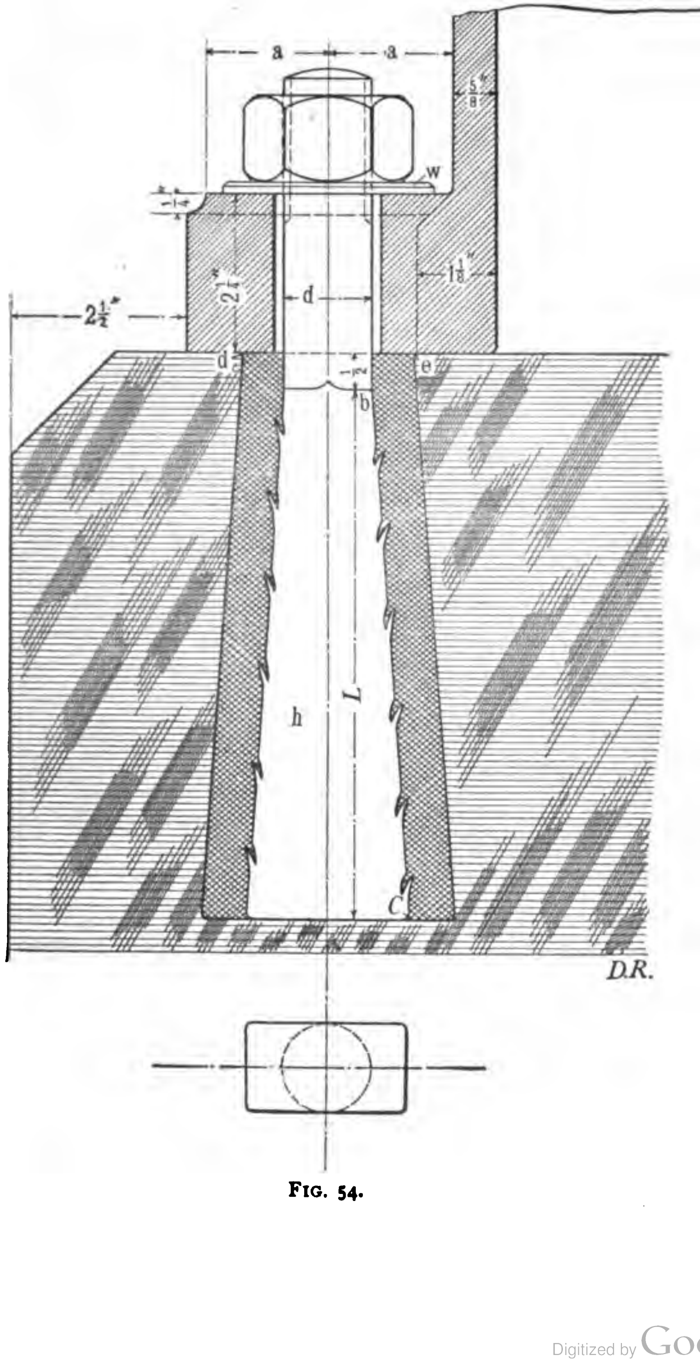


FIG. 54.

purposely made rough and jagged, which obviously increases the resistance the bolt offers against being withdrawn from the hole. The length L of the head (h) is usually made equal to $6d$ and has a taper = $1\frac{1}{2}''$ per foot.

Exercise 15.—Draw a rag-bolt in elevation and plan with a part of a cast-iron engine-frame as shown in Fig. 54, making (d) = $1\frac{1}{4}''$ in diameter. *Scale full size.*

Construction.—Draw the centre line and the line representing the top of the stone foundation, then mark off to (b) the distance which the beginning of the head is below the level of the top of the foundation, and from the point (b) find the taper on one side of the axis in the same manner as in Exercise 13. Make the top of the hole de $\frac{1}{4}''$ greater than the large end of the bolt-head, and through (e) draw a line parallel to the side of the bolt-head bc, which will represent the edge of the hole. To complete the other side of the bolt-head mark off with the dividers equal distances on the other side of the centre line.

The Lewis Bolt, shown in Fig. 55, is used, in some cases, in preference to the rag-bolt, because it can be much more easily removed, which is accomplished by withdrawing the key K. The side bc of the bolt-head (h) has a taper of $1\frac{1}{4}''$ per foot, while the opposite side is parallel to the axis of the bolt. The length L of the head may be made as in the design of the rag-bolt, equal to $6d$.

In Fig. 55, the bolt is shown holding down the pedestal shown in Fig. 54, page 79. The hole that the bolt passes through is rectangular, to allow the pedestal to move laterally. The proportions of the washer are the same as in the last exercise. The thickness (t) of the key is made

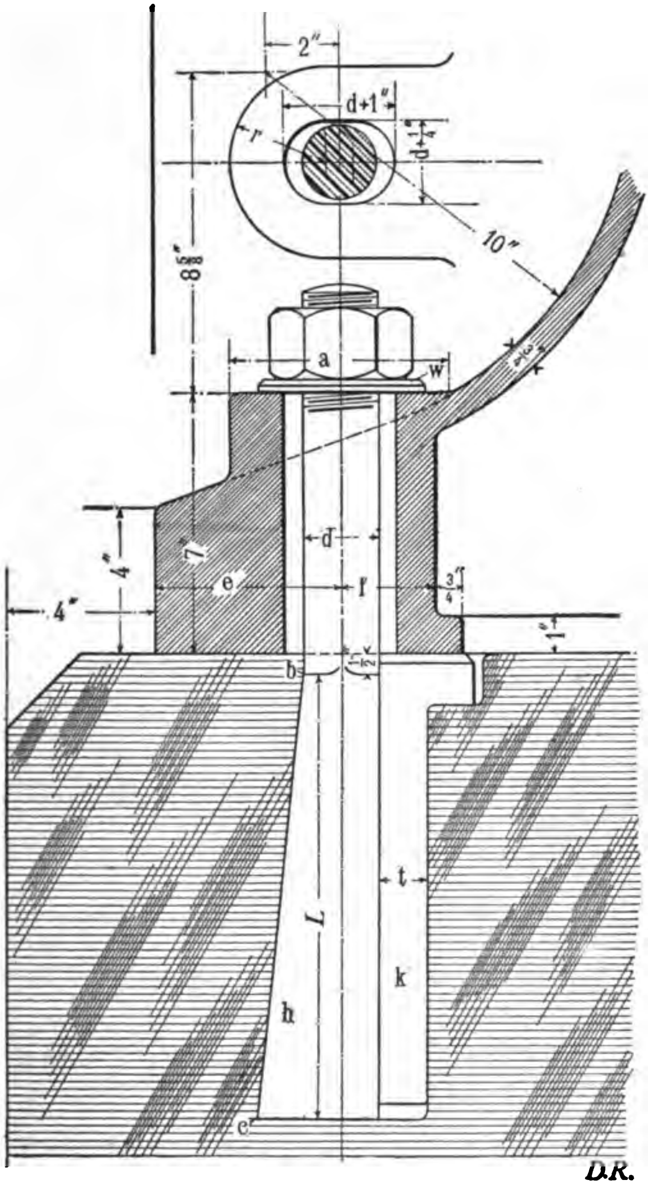

2"
d+1"
r
$d+\frac{1}{4}$"
10"
$8\frac{5}{8}$"
a
w
$\frac{3}{8}$
7"
4"
e
d
f
$\frac{3}{4}$"
4"
1"
b
$\frac{1}{2}$
t
L
k
h
c
D.R.

FIG. 55.

sufficient to allow the large end of the bolt-head to pass through the small end of the hole $+ \frac{1}{4}''$ for clearance, and the point should stop from $\frac{1}{4}''$ to $\frac{1}{2}''$ up from the bottom of the hole. The length of the key-head is made equal to $2t$, and its thickness equal to t.

Exercise 16.—Draw an ELEVATION of a Lewis bolt showing the method of securing it to the foundation, a section of pedestal-base and a PLAN showing the shape of the hole through which the bolt passes, as in Fig. 55. Draw also an END VIEW of the bolt leaving out the foundation-stone and pedestal-base. Make $d = 2''$ in diameter. *Scale half size.*

Construction.—Proceed in the same manner as in the previous exercise. The distance (a) in this case should be equal to the diameter of the washer (w) + the longitudinal movement $+ \frac{1}{2}''$. Make $e = \frac{a}{2} + 2''$, $f = d +$ half the longitudinal movement, $r = \frac{w}{2} + \frac{1}{4}''$.

Anchor-bolts passing through the foundation are recommended in preference to the rag or Lewis bolts wherever it is possible to use them. The heads are made removable, so that the bolts can be inserted from the top, and are either under the foundation or in a recess on the side, as in Fig. 56. The simplest form of removable head is made by screwing a nut upon the lower end after the bolt is in position and driving a split pin through it to keep it from working loose. The objection to this form of head, however, is that the nut cannot be removed without difficulty after it has been in place long enough to rust. The usual and most suitable form of removable head for this class of foundation-bolt is

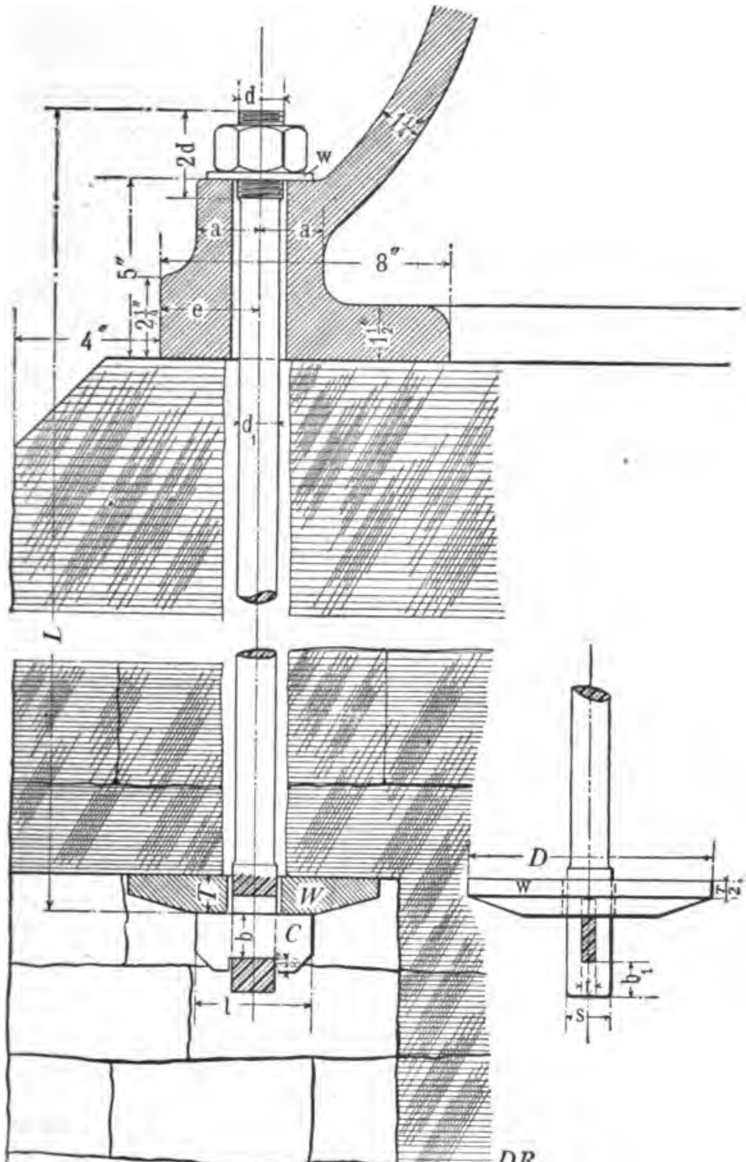


Fig. 56.

shown in Fig. 56. In this design the head end is made square in section, and has a rectangular hole into which the cotter C is fitted. The bolt is kept from turning when the nut is being screwed up by the square end fitting into a corresponding hole in the washer W. To keep the cotter C from working out of place it is provided with gib-heads at the ends. As the strength of a bolt in tension is due to the area at the bottom of the thread, the body of the bolt may be reduced to this extent without reducing its strength.

The proportions of the cotter and the bolt-end through which the cotter passes are

$$b = 1.44d_1, \quad t = .363d_1, \quad S = 1.08d_1,$$

b for shear would $= \frac{d_1}{2}$, but owing to the uncertainty of the longitudinal shearing resistance of the material, it is usual in practice to make it equal S, which insures ample strength. The length l of the cotter should not be less than $2S + \frac{1}{4}''$ and is usually made $= 3S$, which gives a better support to the washer W. The washer W is usually made round or square. When round, D, the diameter, will be found by the formula

$$\frac{\pi D^2}{4} f_c = \frac{\pi d_1^2}{4} f_t, \quad . \quad . \quad . \quad . \quad . \quad (12)$$

from which

$$D = \sqrt{\frac{d_1^2 f_t}{f_c}}.$$

Take the values for f_c, in tons, from the table on page 40, and allow a factor of safety of 20. Take the value of f_t

in this case = 7 tons per square inch. The thickness T of the washer is made from d_1 to $1\frac{1}{4}d$.

Exercise 17.—Draw an ELEVATION of an anchor-bolt for securing an engine-frame to a stone foundation showing the frame-foot and stonework in section; the top stone of sandstone and the under part of brick. Make also an END VIEW of the bolt-head with the cotter in section. Make $d = 1\frac{1}{4}''$, $L = 6'\ 0''$, $a = d + \frac{1}{8}''$, $e = 2d + \frac{1}{4}''$. *Scale half size.*

Cap-screws, the different forms of which are shown in Fig. 57, are employed like the tap-bolt for screwing two or more pieces together. The reason for the name "cap-screws" is that they are used for fastening on caps or covers on machinery, such as the caps of journal-bearings, etc.

The Length of a cap-screw is the distance under the head, excepting the flat-headed form, which includes the thickness of the head in the length. The angle of the cone of the flat-headed screw is about 76°, the sides making angles of 52° with the top, but it is usual to represent the heads on the drawings with the sides making an angle of 60° with the top.

The height of the flat-headed screw is = .7 of the screw diameter. The height of the button-headed screw = .6 of the screw diameter. The width of the saw-cuts on the heads are = .2 of the screw diameter. The other proportions are given in Table 9.

Collar-screws are used for the same purpose as cap-screws. The collar under the head, Fig. 58, gives a larger bearing-surface for the head and is used where the hole through the connected piece is larger than the screw diameter.

TABLE 9.

CAP-SCREWS.

(Worcester Screw Co.)

Hexagon and Square Heads.

Diameter of screw	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1	1⅛	1¼
Threads to inch	20	18	16	14	12	12	11	10	9	8	7	7

Hexagon Head.

Diameter of head	7/16	½	9/16	⅝	¾	13/16	⅞	1	1⅛	1¼	1⅜	1½
Length of head	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1	1⅛	1¼

Square Head.

Diameter of head	⅜	7/16	½	9/16	⅝	11/16	¾	⅞	1⅛	1¼	1⅜	1½	1⅝
Length of head	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1	1⅛	1¼	1⅜

Flat, Round, Fillister, and Button Heads.

Diameter of screw	⅛	3/16	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1
Threads to inch	40	24	20	18	16	14	12	12	11	10	9	8

Flat Head.

Diameter of head	¼	⅜	15/32	⅝	¾	13/16	⅞	1	1⅛	1⅜

Round and Fillister Heads.

Diameter of head	3/16	¼	⅜	7/16	9/16	⅝	¾	13/16	⅞	1	1⅛	1¼
Length of head	⅛	3/16	¼	5/16	⅜	7/16	½	9/16	⅝	¾	⅞	1

Button Head.

Diameter of head	7/32 Full	5/16	7/16	9/16	⅝	¾	13/16	15/16	1	1¼

CAP SCREWS.

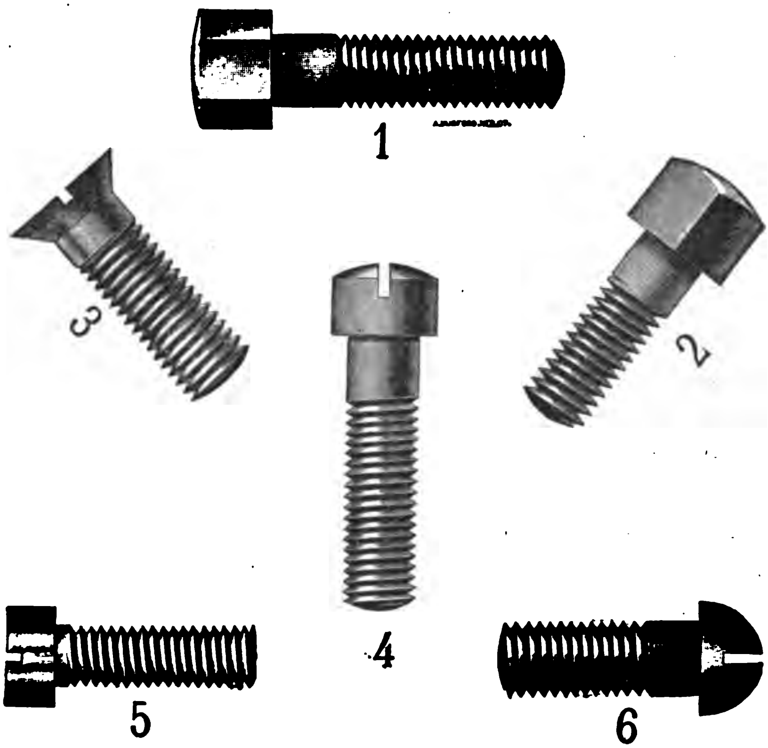


FIG. 57.

FIG. 58.

Set-screws are employed to hold parts of machines in place by setting the point of the screw against the object to be held.

The Holding Power of Set-screws.—In tests (made by G. Lansa, A.S.M.E.) of the holding power of set-screws for securing pulleys to steel shafts by means of wrought-iron screws with points of the form shown at Nos. 1, 2, and 3 (Fig. 59). It was found that the round-pointed form (No. 1), with the radius of the point equal to about the diameter of the screw, had the greatest holding power. The cup-point (No. 2), which was case-hardened, held well while the edges were sharp, but the holding power decreased after the first test because of the edges becoming flat. This serves to show that to get good results with this form of point the screw must be made of a harder material than that of the piece it is holding, and should not be used where the point is subjected to excessive wear.

The length of the head and the distance across the flats are equal to the diameter of the screw. The diameters of the round and flat pivot-points (No. 8) are equal to the diameter at the bottom of the threads, and the length of point = from $.5d$ to d. The angle of the cone and hanger set-points is usually $45°$ or $60°$.

Strength of Bolts.—In an ordinary bolt with a V thread employed for holding two or more pieces together by the pressure due to the screwing up of the nut, the bolt would yield (1) by tension combined with a torsional stress due to the friction between the threads of the nut and those of the bolt. This combination of tension and torsion causes the bolt to part where the thread ends, because of the rapid change of section; (2) by shearing off the threads; (3) by shearing off the bolt-head. Comparing (1) with (2) it will be found that the shearing strength of the thread on the nut is

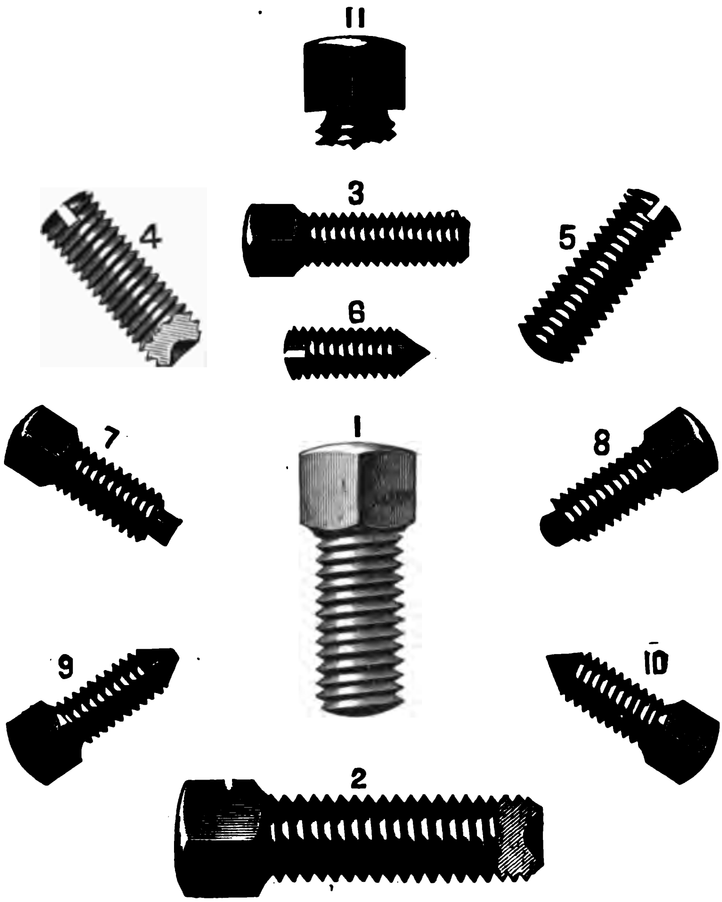


FIG. 59.

SET-SCREWS.

No. 1. Regular round point, set.
" 2. Cup point, set.
" 3. Flat " "
" 4. Cup " headless.
" 5. Round point, headless.
" 6. Cone " "

No. 7. Flat, pivot point.
" 8. Round, " "
" 9. Hanger, set "
" 10. Cone point.
" 11. Necked style.

Diameter of screw	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1 1/8	1 1/4
Threads to inch	20	18	16	14	12	12	11	10	9	8	7	7

equal to about twice the strength of the section at the bottom of the thread, but in practice it is found that when the depth of the nut is made less than .7 of the bolt diameter, the threads are injured. Bolts or studs used for face-joints on vessels subjected to internal pressure, depend upon the care exercised by the workman to leave sufficient strength to withstand the pressure after the bolt is screwed up. As the amount of strength left is an unknown and uncertain quantity, the stress upon the bolts calculated from the internal pressure should be kept very low, and no face-joints, unless very small ones, should have bolts less than $\frac{5}{8}''$ in diameter. In permanent joints the stress thus calculated per square inch of section of bolt at the bottom of thread should not exceed 6000 lbs.; and for bolts in joints frequently broken the stress should be as low as 2000 lbs. Thus d_1, the diameter at the bottom of the thread, to withstand the required pressure, will be found by the formula

$$d_1 = \sqrt{\frac{p \times a}{.7854 \times nf_t}}, \quad . \quad . \quad . \quad . \quad . \quad (13)$$

a = area of exposed surface in square inches;

p = the pressure per square inch;

n = the number of bolts;

f_t = the strain per square inch;

Unwin's formula for cylinder-bolts or studs is

$$d_1 = \sqrt{\left(\frac{p}{nf}\right)}D, \quad . \quad . \quad . \quad . \quad . \quad (14)$$

$D =$ the diameter of the cylinder;
$p =$ the pressure per square inch;
$n =$ the number of bolts.
$f =$ the strain per square inch $= 2000$ lbs. when the diameter of the cylinder is $10''$ or less, and 4000 lbs. when above.

Cylinder-cover and steam-chest cover-bolts should be of soft steel.

Bolts of Uniform Strength.—When a bolt in tension is subjected to irregular strains and heavy vibrations, it is made lighter and stronger by making the area of the cross-section of the unscrewed part equal to that of the screwed part at the bottom of the threads. This is obtained by turning down the bolt-body to the same diameter as the screwed part at the bottom of the threads, leaving a part at each end to fit the hole, as shown in Fig. 60.

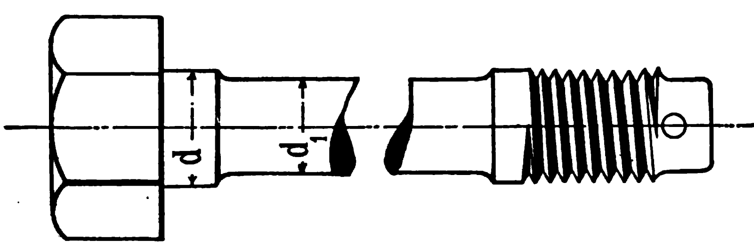


FIG. 60.

Another method adopted where it is necessary that the bolt should fill the hole it is fitted into, is to drill a hole through the centre of the bolt from the head up to where the screw ends, as shown in Fig. 61.

The diameter of the hole is found by the formula

$$d_2 = \sqrt{d^2 - d_1^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where d_2 = diameter of the bolt;

d = outside diameter of the bolt-body;

d_1 = the diameter at the bottom of the thread.

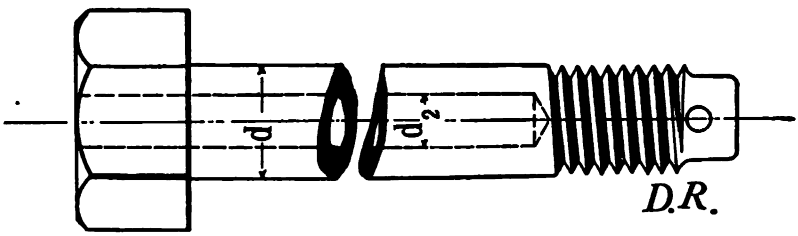


FIG. 61.

Nut-locking Devices.—The pitch of the threads on screw fastenings is such that nuts subjected to constant pressure will not slack back because of the frictional holding power between the threads of the nut and those of the bolt combined with the friction between the bearing-surface of the nut and the piece it is fastening. If, however, the pressure is intermittent and there is much vibration, the nut will slack back when the load on it has been sufficiently reduced to allow the vibrations to overcome the friction which opposes the turning of the nut. Consequently, wherever a screw is subjected to much vibration and a varying load, the nut will gradually slack back and allow the connection to work loose unless some locking device is used to keep the nut from rotating backward.

A Jam-nut is the simplest and most frequently employed device. This is simply a second nut N (Fig. 62) screwed down on the top of the lower nut L as tightly as possible, and the lower nut turned back to cause the threads in the nut N to press upon the under side of the threads on the bolt, while the threads in the nut L press upon the upper

side of the bolt-threads. Hence all slack between the threads of the bolt and those of the nuts is taken up and the nuts will have a frictional holding power independent of the

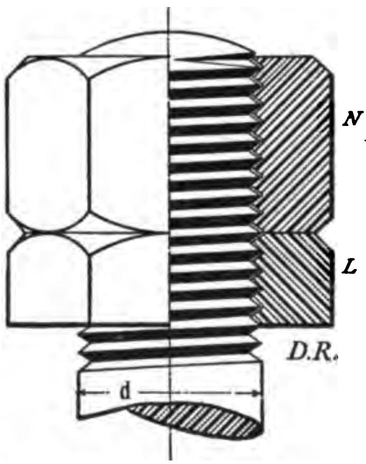


FIG. 62.

tension on the bolt. By this arrangement the load on the bolt is carried on the upper nut, which should be the larger. In practice, however, the thin nut is often put on the top because when screwed down first it requires a special spanner to turn it without disturbing the upper nut, the ordinary spanner or wrench being too thick. The general rule is to make the thin nut equal to half the diameter of the bolt, but many engineers use two ordinary nuts, thus making the height of the nuts equal to twice the diameter of the screw. Others again make a compromise between these methods and make the height of each nut equal to $\frac{3}{4}$ of the screw diameter. We recommend the latter method and have used these proportions wherever jam-nuts are shown. This method of

locking is too cumbersome to be used on large-sized nuts. It is rarely employed on nuts over $1\frac{1}{4}''$ in diameter.

Spring-washer Nut-lock.—This consists of a single coil of a steel spring, *NL*, Fig. 63, which keeps the nut *N* from

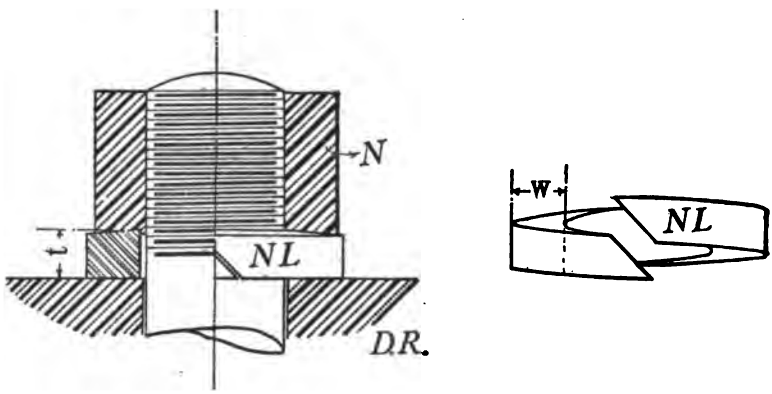


FIG. 63.

slacking back, by its elasticity, when the tension on the bolt is reduced. It is employed quite extensively in railway-engineering practice for securing nuts subjected to the heavy vibrations common to this class of work. The form shown in Fig. 63 is that made by the American Brake Beam Co., and is employed to secure the nuts on the bogie frames, etc., manufactured by them. In the cross-section the top of the washer is inclined $\frac{1}{32}$ of an inch, and when the nut is screwed home its under side conforms to the part of the washer in contact with it. The following proportions agree approximately with the washers manufactured by the afore-mentioned company:

The outside diameter $= F$ the distance across the flats of the nut $+ \frac{1}{16}''$.

The inside diameter = d the diameter of the bolt + $\frac{1}{8}''$.

The mean thickness t is equal to the width w.

Exercise 18.—Draw an elevation of a spring-washer nut-lock before the nut is screwed down, as shown in Fig. 63. Make $d = 1''$ diameter. *Scale twice full size.*

Wiles's Nut-lock, shown in Fig. 64, is an ordinary nut sawn half way across. After the nut is screwed home the

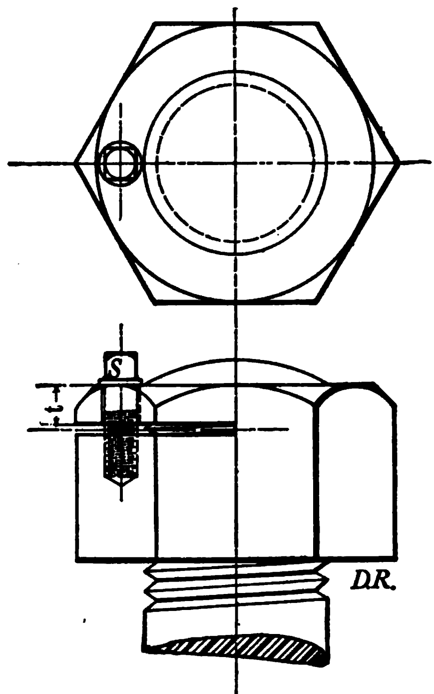


FIG. 64.

opening is partly closed by the screw S, which causes the threads in the upper part of the nut to clamp the corresponding threads of the bolt. The thickness t of the clamping

part of the nut may be made equal to twice the pitch of the threads. The diameter of the screw $S = \frac{F}{2} - \frac{d}{2}$, where $F =$ distance across the flat sides of the nut, and d nominal diameter of bolt. As there is not sufficient room on nuts under 1″ in diameter to use a set-screw, they are locked by partly closing the saw-cut with a hammer blow before the nut is put upon its screw.

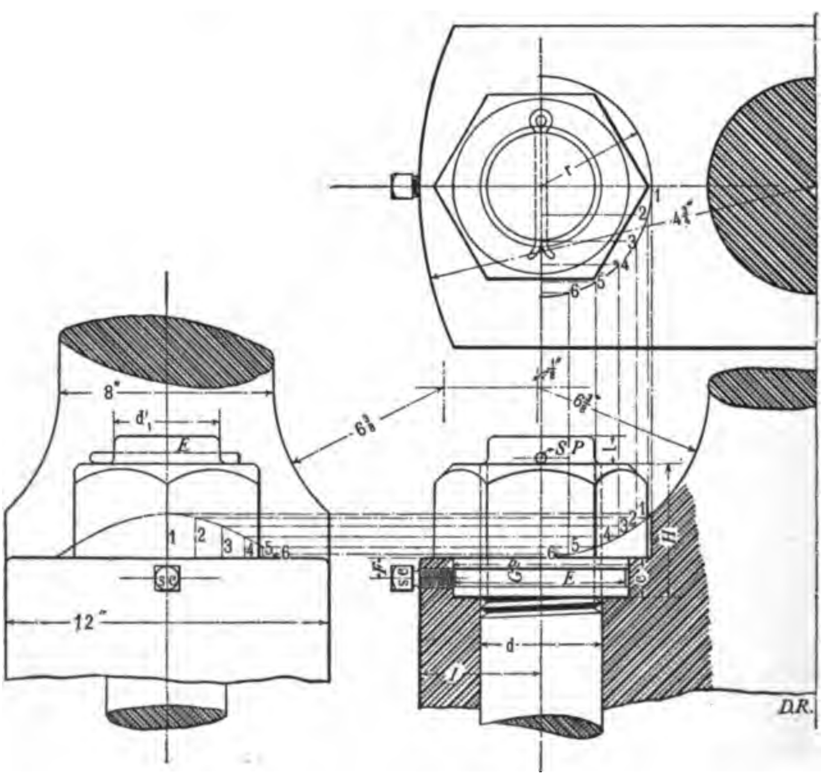


Fig. 65.

Nuts Locked by Means of Set-screws.—The arrangements shown in Fig. 65 are used on quick-moving parts of

machines. They are neat in appearance, simple, and effective when subjected to the worst conditions. In Fig. 65 the lower part of the nut is turned to form a cylindrical projection which fits into a corresponding counterbore in one of the pieces connected by the bolt. Through the latter passes a set-screw S, the point of which presses on the bottom of the groove cut upon the cylindrical projection, to keep the burs raised by the set-screw from interfering with the nut being removed.

The following proportions agree with general practice:

$H = d + \frac{1}{4}''$;
$S = \frac{1}{8}d + \frac{1}{8}''$;
$C = 2S = \frac{1}{4}d + \frac{1}{4}''$;
$F = S$;
$J = d$;
$G =$ diameter of set screw at the bottom of the threads;
$D = 1\frac{1}{2}d - \frac{1}{8}''$;
$E = 1\frac{1}{2}d$;
$r = d$.

In addition to the locking device it is usual, on quick-moving parts, to extend the bolt beyond the nut. This extension E, called a pin-point, has the threads cut off and a hole drilled through it into which is fitted a split pin SP. This renders the nut secure against coming off, but does not necessarily prevent its slacking. The diameter of the split pin SP is $.05d + .13$, $l = 2\frac{1}{2}$ times the diameter of the split pin, $d_1 =$ diameter at the bottom of the threads. A method of drawing split pins is shown in Fig. 69.

Exercise 19.—Draw a plan, front elevation and end elevation of the locking arrangement, shown in Fig. 65, showing the application of the arrangement on a connecting-rod end of the form shown in Fig. 66. Make $d = 4\frac{1}{2}''$. *Scale half size.*

Construction.—Locate the centre lines, draw the hexagon and the part of the connecting-rod end in plan. This is as far as we can proceed without the elevation. Draw the part of the connecting-rod in front elevation and complete the locking arrangement, projecting the parts already drawn in the plan view. Taking our measurements from the plan, and projecting from the front elevation, we can complete the end elevation. We can now complete the plan from the front elevation by projecting the parts not already drawn in that view. The method of drawing the curve formed by cutting the fillet to allow the nut to bear upon a flat surface, will be understood by following the construction lines 1, 2, 3, 4, 5, 6. In drawing office practice this curve is usually drawn by an arc of a circle passing through the limiting points. *All parts are dimensioned in inches.*

When it is undesirable to counterbore the piece upon which the nut bears, as in Fig. 65, the cylindrical portion of the nut is made to fit into the collar C, Fig. 66, which is carried upon the outer surface of the connected piece, and kept from rotating by means of the pin P. The nut is secured by the set-screw S passing through the collar and pressing on the bottom of the groove which is cut upon the cylindrical part of the nut. In some cases the pin P is fitted into the hole in the connected piece, but in the example shown in Fig. 66 it is screwed into the piece to avoid the risk of losing it when the nut and collar are removed. The proportions of the nut are the same as in the last exercise. The collar proportions are $D = 2d$, $C = 2S = \frac{1}{4}d + \frac{1}{4}''$. The diameter of $P = \frac{1}{8}d + \frac{1}{16}''$. The length of $P = 3$ times its diameter, half of which fits into the collar.

Exercise 20.—Draw an elevation of the locking arrangement shown in Fig. 66. Make $d = 2''$. *Scale full size.*

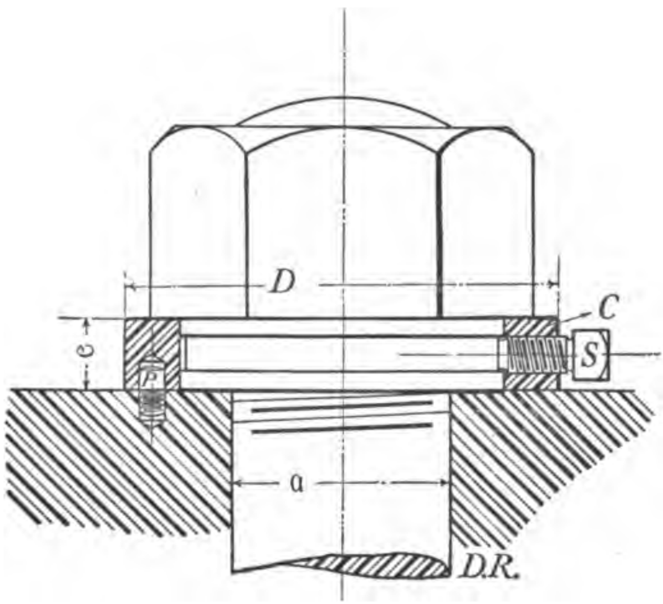


FIG. 66.

Circular Nut-locking Device.—The nut and its locking arrangement shown in Fig. 67 are used for securing the piston-rod to a cross-head of the form shown in Fig. 67. On the outer surface of the nut *N*, longitudinal grooves are cut, into which the projections on the spanner, employed for screwing it up, fit. The locking-plate *LP* consists of a plate shaped to suit the curvature of the nut, and has a projection which fits into one of the spanner grooves. The stud *S* is screwed into the surface upon which the nut is carried, passing through the groove in the locking-plate (*LP*) and is prevented from unscrewing by making the part within the

locking-plate square. The method of locking the nut is as follows: The stud S is screwed into the metal at the proper

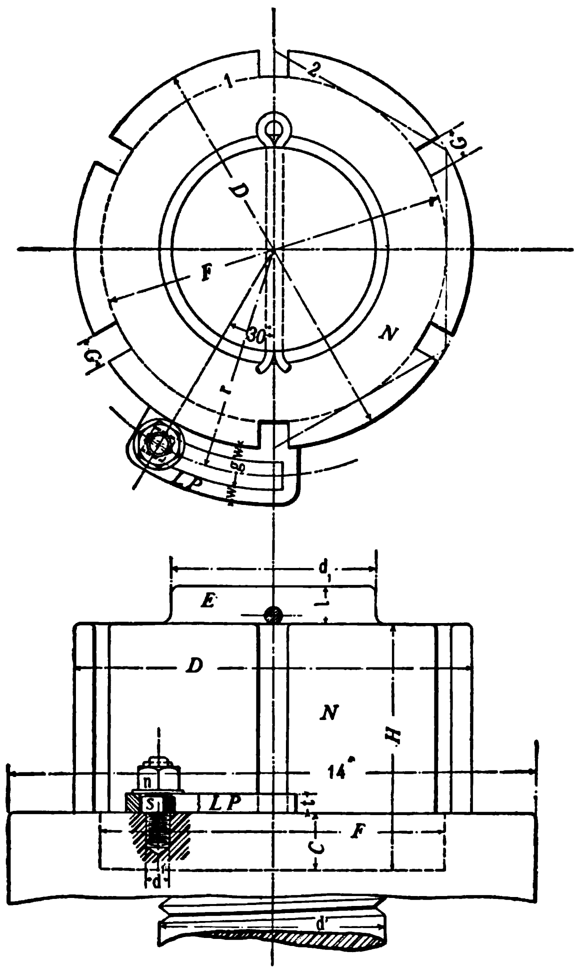


FIG. 67.

distance from the centre of the nut N, forming an angle with the radial line which passes through the centre of the projec-

tion on the locking-plate equal to half the angular distance between two of the spanner slots. This allows the nut to be locked in any position. After the nut N is screwed into position, the locking-plate LP, with its projection fitting into one of the grooves, is passed over the stud until it rests upon the piece fastened by the nut N. The nut can then be locked securely by clamping the locking-plate LP, by screwing down N. The nut N has a cylindrical projection on the under side which fits into a corresponding recess in the piece upon which it bears. This insures that the outside of the nut is concentric with the arc which passes through the centre of the locking-plate. It also gives a greater length of nut without increasing the distance which the nut projects from the piece it is fastening.

The proportions of the nut and its locking device are as follows:

$D =$ diameter across the angles of the hexagon;
$F =$ diameter across the flats of the hexagon;
$H = d + 1''$;
$C = \frac{1}{4}d$;
$G = \frac{1}{8}d$;
$t = \frac{1}{10}d$;
$W = .04d + .13$.

Make the diameter d' of the stud $S = \frac{1}{2}''$ when nut N is $2''$ or under, and $\frac{5}{8}''$ for all nuts over $2''$ in diameter. The width of the groove in the locking-plate equal to the diameter of the stud $+ \frac{1}{16}''$. The width of the square body on the stud $= d'$ and its length $= t - \frac{1}{8}''$.

Exercise 21.—Draw a fluted circular nut and locking

arrangement, as shown in Fig. 67. Make $d = 6''$. *Scale* 8″ *to the foot.*

Construction.—Locate the centre lines, draw the circle making the diameter $F = 1\frac{1}{2}d + \frac{1}{8}''$. Tangent to the circle draw the line 2 to make an angle of 30° with the horizontal. Determine the radius r of the arc passing through the centre of the nut-lock and complete the plan of the nut-lock. Determine the centres of the spanner grooves by circumscribing on the half of the circle 1, three sides of a hexagon, as in Fig. 67. The sides of the groove are parallel to the radial lines which bisect the angles formed by the sides of the hexagon. Projecting from the plan complete the elevation. *Construction lines are not to be inked in.*

In Fig. 68 is shown a nut-locking device used for securing the piston-rod to the piston shown in Fig. 66. The nut N in this case is of cast steel and has a projection on the under side which fits into a corresponding recess in the locking-plate LP, which in turn fits into a circular recess on the piston. The locking-plate has a tapped hole through it, and through this tapped hole, at right angles to its axis, the ring is cut. After the nut with its locking-plate has been screwed into place a tapered plug P is inserted into the tapped hole. This opens the saw-cut and forces the locking-plate against the sides of the circular recess on the piston. The nut is thus securely locked by the friction caused by the pressure of the locking-plate against the sides of the recess. The following proportions may be used for the nut and locking-plate:

d = nominal diameter of screw;

F = distance across the flats—$\frac{1}{8}''$;

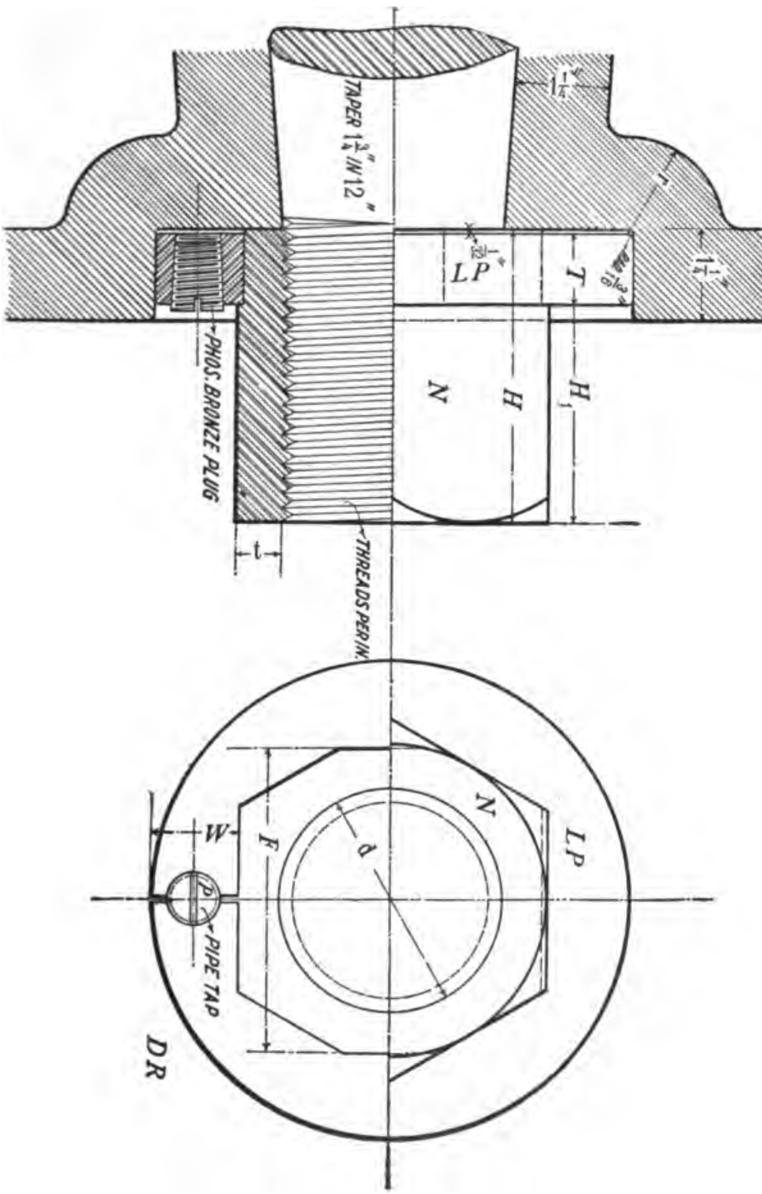

TAPER $1\frac{3}{4}''$ IN $12''$
$1\frac{1}{4}''$
r
LP
T
H
H_1
N
$1\frac{1}{4}''$
PHOS. BRONZE PLUG
t
THREADS PER IN.
W
F
d
N
LP
P
PIPE TAP
DR

FIG. 68.

$t =$ thickness of standard nut having the same number of threads per inch;

$H =$ diameter of screw + thickness of locking-plate;

$T = .09d + .7$.

The size of the pipe-tap is $= \frac{1}{8}d$ and $\frac{3}{8}''$ pipe-tap where $d = 3''$ or more. The projection on the under side of the nut $= T + \frac{1}{32}''$ to allow the nut to bear upon the piston. $W =$ twice the diameter of the tapped hole at the small end.

Exercise 22.—Draw the nut-locking arrangement shown in Fig. 68, showing part of the piston and piston-rod. Make $d = 3''$ and having 5 threads per inch. *Scale full size.*

Construction.—To find the distance across the flats of the hexagon turn to Table 8, page 72, and find the thickness of a nut having 5 threads per inch by subtracting the radius of the screw from half the distance across the flats. To find the diameter of the tapped hole at the small end, turn to the table of Wrought-iron Pipes on page 57. The size of the actual outside diameter is the diameter of the tapped hole at the large end, and the hole is $\frac{1}{16}''$ less in diameter for every $1''$ of its length. Complete the drawing, substituting the dimensions in inches for the reference letters, and give the number of threads per inch on the piston-rod screw and the nominal diameter of the pipe-tap.

Pin and Pin-joints.—Pins connect pieces by their resistance to shearing at one or two cross-sections.

Split Pins, when made of a uniform diameter from wire of a semicircular cross-section and provided with a head, as in Fig. 69, are used for preventing pieces from separating, while allowing a slight motion in the direction of the axis of the piece that they pass through, as in Fig. 67.

The method of drawing split pins is clearly shown in Fig. 69. The diameter of the pin, in proportion to the diameter d of

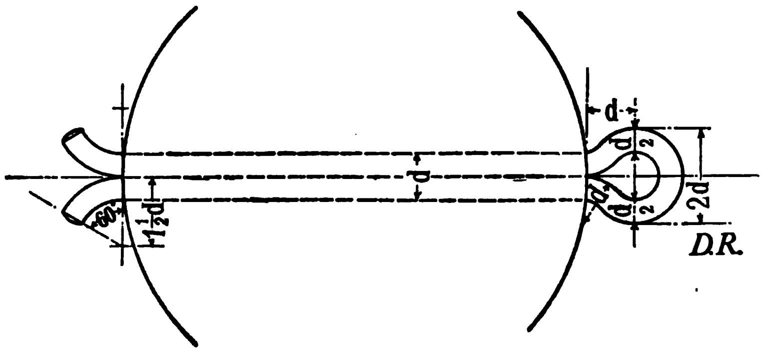


FIG. 69.

the piece it passes through, may be $= .05d + .13$, taking the nearest size in $\frac{1}{16}''$.

Taper Pins, shown in Fig. 70, are used for securing one piece to another in a fixed position, as shown in Fig. 71.

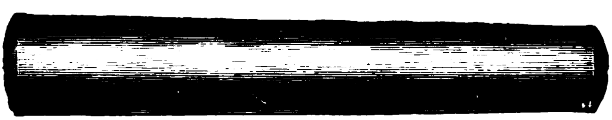


FIG. 70.

They are sometimes split at the small end, and opened out in the same manner as the ordinary split pin, to prevent slacking back. The diameter of the tapered pin at the large end, in proportion to the diameter (d) of the piece through which it passes, may be made $= .06d + .13$ and taking the nearest size from Table 10.

TABLE 10.

STANDARD STEEL TAPER-PINS.

Taper one-quarter inch to the foot.

Number........	0	1	2	3	4	5	6	7	8	9	10
Diameter at large end	.156	.172	.193	.219	.250	.289	.341	.409	.492	.591	.706
Approximate fractional sizes	5/32	11/64	3/16	7/32	¼	19/64	11/32	13/32	½	19/32	23/32
Longest limit of length	1	1¼	1½	1¾	2	2¼	3¼	3¾	4½	5¼	6

A **Knuckle-joint** is a pin-joint used for connecting two rods in such a manner that one of them will have a rotary

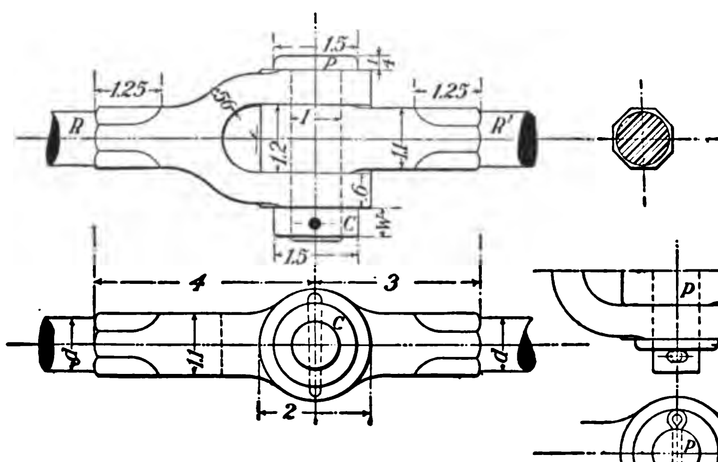


FIG. 71.

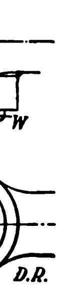


FIG. 72.

motion in one plane. The connection is made, as shown in Fig. 71, by the pin *P* passing through the fork, or double eye, formed on the rod *R*, and the single eye, on the rod *R'*,

which fits into the fork. The parts of the rods near the eye and fork are either left square or have the corners taken off for a distance, which makes a part of the rod octagonal in cross-section. In the arrangement shown in Fig. 71, the pin *P* is allowed to turn and is kept in place by the collar *C*, which is secured to the turning-pin *P* by driving a taper-pin through it and the collar. The width *W* of the collar should not be less than 2½ times the diameter of the taper-pin.

Another method in common use for holding the turning-pin in place is to use a loose washer (*W*) and split pin, as shown in Fig. 72. In Fig. 73, the pin *P* is held against

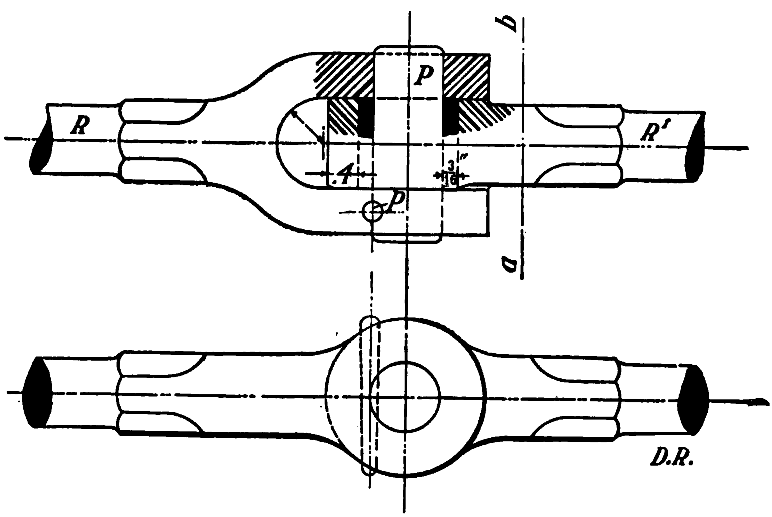


FIG. 73.

turning by a taper-pin *p* driven transversely through one of the eyes on the rod *R* and partly into the pin *P*. By this arrangement all the wear, due to the turning motion, is on the eye of the rod *R'*, which is fitted with a steel or bronze bush.

The Proportions given in Figs. 72 and 73 make the joint stronger than the solid rod. This is necessary to allow for bending stresses produced when the pin becomes worn. *Unit of proportions d.*

Exercise 23.—Draw a PLAN, ELEVATION, and END VIEW of the joint shown in Fig. 71, showing the method of holding the pin in place by means of a split pin and washer. Make $d = 1''$ *Scale full size.*

Exercise 24.—Draw a PLAN partly in section, an ELEVATION and SECTIONAL END VIEW (the plane of section passing through the rod at the line *ab*) of the knuckle joint shown in Fig. 73. Make $d = 1\frac{1}{2}''$. *Scale full size.*

CHAPTER II.

KEYS, COTTERS, AND GIBS.

Keys are employed to connect wheels, cranks, cams, etc., to shafting transmitting motion by rotation. They are generally made of wrought iron or steel, and are commonly rectangular, square, or round in cross-section. The form of key in general use is made slightly tapered and fits accurately into the key-way, offering a frictional holding power against the keyed piece moving along the shaft. The groove or part where the key fits on the shaft, and the groove into which it fits on the piece it is holding is called the key-bed, key-way or key-seat. For square or rectangular keys, when the keyed piece is stationary on the shaft, the bottom of the groove on the shaft is parallel to the axis, while that of the groove in the piece it is securing is deeper at the one end than the other to accommodate the taper of the key.

Keys may be divided into three classes: 1. Concave or saddle key; 2. flat key; 3. sunk key.

Saddle Key.—This form of key has parallel sides, but is slightly tapered in thickness and is concaved on the under side to suit the shaft, as shown in Fig. 74. As the holding power depends entirely upon the frictional resistance, due to the pressure of the key on the shaft, the saddle key is only

adapted for securing pieces subjected to a light strain. When this key is used for securing a piece permanently, the taper is usually made 1 in 96, but when employed on a piece requiring to be adjusted, such as an eccentric, the taper is increased to 1 in 64 to allow the key to be more easily loosened.

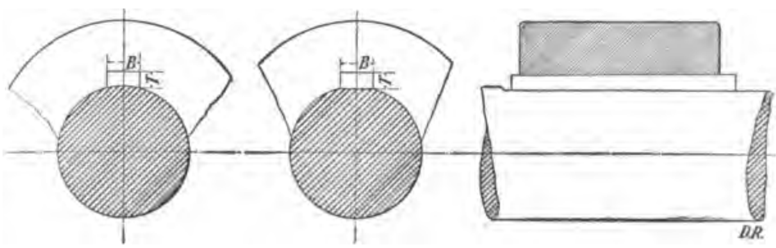


FIG. 74. FIG. 75.

Flat Key.—This form of key, Fig. 75, differs from the saddle key in that it rests on a flat surface filed upon the shaft. It makes a fairly efficient fastening, but as it drives by resisting the turning of the shaft under it, there is a tendency to burst the keyed-on piece.

TABLE 11.

DIMENSIONS OF SADDLE AND FLAT KEYS.

D	1	1¼	1½	1¾	2	2½	3	3½	4	5	6	7	8
B	¼	5/16	⅜	7/16	½	⅝	¾	⅞	1	1⅛	1⅜	1½	1¾
T	3/16	3/16	3/16	¼	¼	5/16	5/16	⅜	⅜	7/16	½	9/16	⅝

Sunk Keys are so called because they are sunk into the shaft and the keyed-on piece, Fig. 76, which entirely prevents slipping. For engine construction they are usually rectangular in cross-section and made to fit the key-seat on all sides. When subjected to strains suddenly applied, and

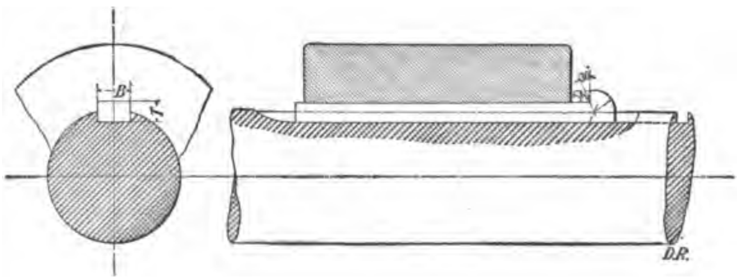


FIG. 76.

in one direction, they are placed to drive as a strut, diagonally, as in Fig. 77.

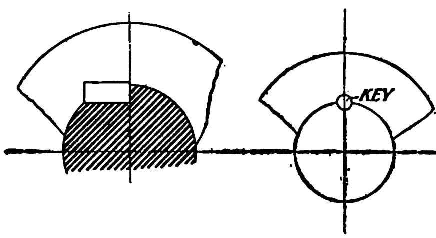


FIG. 77.

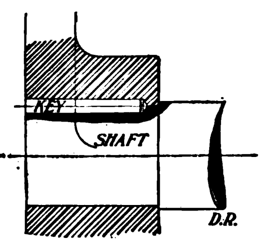


FIG. 78.

The following table, taken from Richards's "Machine Construction," agrees approximately with average practice:

TABLE 12.

DIMENSIONS OF RECTANGULAR SUNK KEYS.

D	1	1¼	1½	1¾	2	2½	3	3½	4	5	6	7	8
B	¼	5/16	⅜	7/16	½	⅝	¾	⅞	1	1⅛	1⅜	1½	1¾
T	5/32	3/16	¼	9/32	5/16	⅜	7/16	½	⅝	11/16	13/16	⅞	1

In mill-work, for fastening pulleys, gear-wheels, couplings, etc., to shafting they are made slightly greater in depth

than breadth. For machine tools they are generally square in cross-section. The following table gives the sizes of keys used by Wm. Sellers & Co. both for shafting and machine tools:

TABLE 13.

	"	"	"	"	"	"	"	"	"
D	1½	1¾	2	2¼	2½	2¾	3	3¼	3½
B	5/16	5/16	7/16	7/16	9/16	11/16	11/16	11/16	11/16
T	⅜	⅜	½	½	⅝	¾	¾	¾	¾
	"	"	"	"	"	"	"	"	"
D	4	4½	5	5½	6	6½	7	7½	8
B	13/16	13/16	13/16	15/16	15/16	15/16	1 1/16	1 1/16	1 1/16
T	⅞	⅞	⅞	1	1	1	1⅛	1⅛	1⅛

Round Keys.—Taper-pins (Fig. 78) are sometimes used as keys to prevent rotation where a crank or wheel is shrunk on to the end of a shaft or axle. Round keys are used in such a case because of the ease in forming the key-way, which is simply a tapered round hole drilled half into the shaft and half into the shrunk-on piece. The standard proportions of the pins are given on page 106. The size at the large end nearest to ¼ of the shaft diameter may be used for this purpose.

Fixed Keys are used when it is undesirable to cut a long key-way on the shaft to allow the key to be driven into place after the keyed-on piece is in position. The fixed key is sunk into the shaft, as in Fig. 79, and the keyed-on piece is driven into position after the key is in place.

When a keyed-on piece has to be adjusted to different positions on the shaft, to avoid the trouble of drawing a tight key in and out, it is made to slide in the key-way, and the keyed-on piece is held against moving along the shaft by means of set-screws, as shown in Fig. 80.

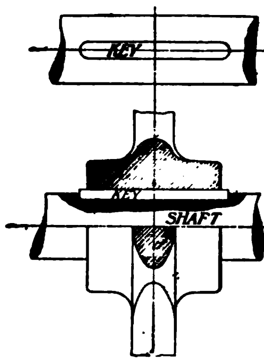


Fig. 79.

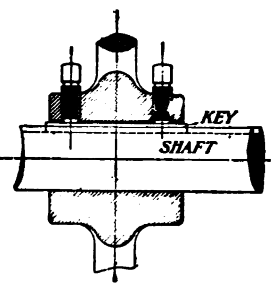


Fig. 80.

Sliding Feather Key.—This system of keying secures the piece to the shaft, to transmit motion of rotation, and at the same time allows the keyed-on piece to move along the

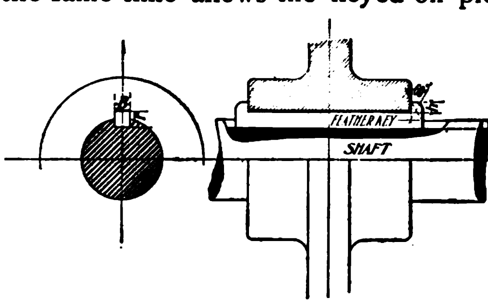


Fig. 81.

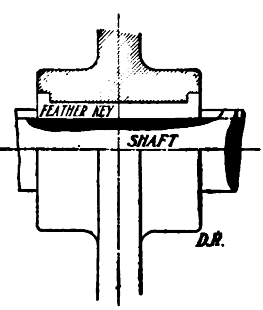


Fig. 82.

shaft. They may be secured to the keyed piece and slide in a groove on the shaft, as in Fig. 81, or secured to the shaft and slide in the groove in the keyed piece, as in Fig. 79. The dimensions for this form of key may be taken from Table 12.

Woodruff Keys.—This system of keying (Fig. 82*a*) is used for machine tools, or wherever accurate work is of first importance. With this form of key, as the key rights itself to the groove in the keyed-on piece, there is no danger of

the work being thrown out of true by badly fitted keys, and, being deep in the shaft, it cannot turn in the key-seat.

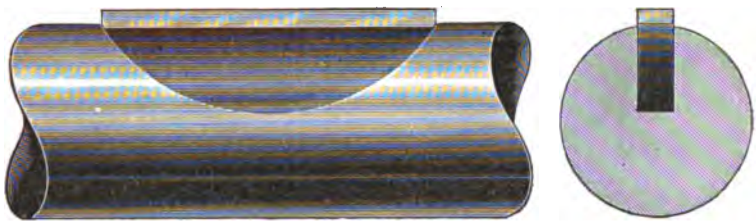


FIG. 82*a*.

Key-heads.—When the point of a key cannot be conveniently reached for the purpose of driving it out, a head is formed on one end, as shown in Fig. 76. Which shows the proportions and method of construction given in RICHARDS'S "MACHINE CONSTRUCTION."

Strength of Keys.—The driving power of saddle keys or keys on flats cannot be calculated with any degree of accuracy. They are used only where the power transmitted by the keyed-on piece is small.

Sunk Keys are subjected to shearing and crushing strains, and are required (1) to transmit the whole of the power transmitted by the shaft, as in crank-shaft couplings, etc., or (2) only a part of the power transmitted by the shaft, as when fastening pulleys, eccentrics, etc. As a general rule, however, all keys are proportioned to suit the first conditions, unless where the amount of power transmitted by the shaft is exceedingly great in comparison with that taken off at the keyed-on piece.

Let B = breadth of key;

L = length of key;

$\frac{d}{2}$ = radius at which key offers a resistance;

the shearing of the material which is = 9000 for wrought iron and 11,000 for steel.

$.190d^3f_s$ = modulus of the section of shaft for torsion = $1720d^3$ for wrought-iron and $2182d^3$ for steel shafts;

R = the radius of arm through which P, the power, is transmitted.

Under the first conditions the strength of a tight key would be found by the formula

$$f_sBL\frac{d}{2} = .196d^3f_s, \quad . \quad . \quad . \quad . \quad . \quad (16)$$

and under the second conditions by the formula

$$f_sBL\frac{d}{2} = PR. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

In the system of sliding keys the crushing action on the key is greater than when the key is a tight fit in the key-way, and keys of this type should be proportioned to have the moment of shaft torsion = the moment of key shearing = moment of key crushing. Then

$$.196d^3f_s = f_sBL\frac{d}{2} = f_c\frac{HL}{2}\frac{d}{2}, \quad . \quad . \quad . \quad (18)$$

and if we take $f_c = 2f_s$, then $H = B$. In practice, however, H is generally greater than B.

Length of Key.—From the aforegoing formulæ it will be seen that the strength of the key is directly proportional to (L) the length. To find the length L when the full power of

the shaft is to be transmitted through the key. From formula No. 17

$$f_s BL\frac{d}{2} = 196d^3 f_s,$$

$$11000BL\frac{d}{2} = 2182d^3,$$

$$BL = \frac{2182d^2}{5500},$$

substituting the value of B from Table 12 in terms of d

$$L = \frac{2182d^2}{5500 \times .25d} = 1.6d.$$

Hence when the shaft and the key are of the same material, the length (L) of the common key (Table 12) should not be less than $1.6d$. When the hub of the keyed-on piece is so short that one key has not sufficient strength, two or more keys are used. Where two keys are used they should be placed at right angles to each other. By this arrangement the keyed-on piece is held upon three points, which prevents it from rocking upon the shaft when the shaft is not a tight fit in the hole.

COTTERS

are keys employed to connect pieces which are subjected to tensile and compressive forces. They are driven transversely through one or both of the connected pieces and transmit power by a resistance to shearing at two cross-sections. The cotters are usually made rectangular in cross-section, and the ends rounded, as shown in Fig. 83.

The cotter-way with the rounding ends is generally adopted, as it is easier to make, which is done by drilling two

holes of a diameter equal to the thickness of the cotter and cutting out the metal between them. Again, this form of cotter-way does not weaken the cottered pieces to quite the same extent as when the corners are left sharp. The cotters, however, are not so easily fitted into cotter-ways with round ends, and for that reason some engineers make the cotters of rectangular cross-section, fitted into corresponding cotter-ways.

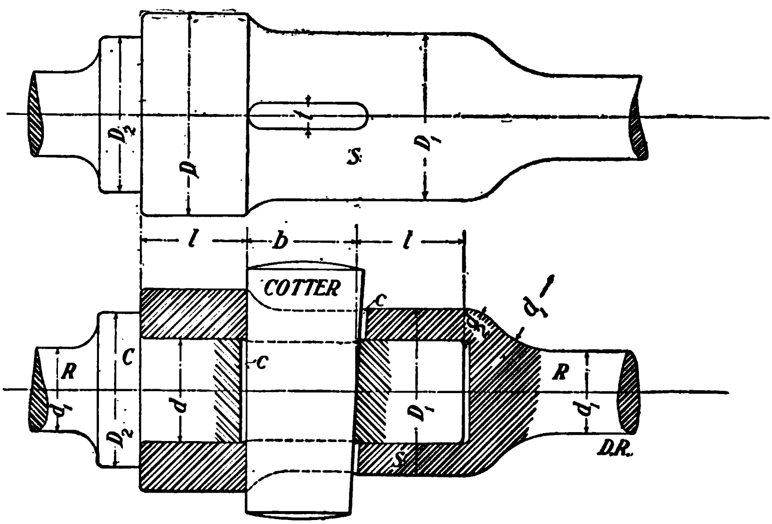


FIG. 83.

Taper of Cotters.—When cotters are employed as a means of adjusting the length of the connected pieces, or for drawing them together, they are made tapered in width, as in Fig. 83, but when used as a holding-piece only, the sides are parallel, as in Fig. 56. When tapered cotters depend upon the friction between their bearing-surfaces for retaining

them in position the taper should not be more than 1 in 24 ($\frac{1}{2}''$ per foot), but where special means are employed for holding the cotter against slacking, the taper may be made as great as 1 in 6 ($2''$ per foot).

Forms and Proportions of Cotter-joints.—When the fastening is subjected to tension only, the arrangement shown in Fig. 84 is used for securing two pieces together by means of a cotter. Fig. 83 shows a method of fastening two rods, R and R', together to resist thrust and tension. The joint is made by fitting the end of the rod R into a socket S formed on the end of the rod R', and through the socket and rod end driving a cotter until the collar C bears against the socket end.

As a cotter-joint is proportioned to withstand the greatest longitudinal force transmitted by the rod, all parts will therefore be proportional to the diameter d_1 of the rod, unless where the dimensions of the rod are increased to insure stiffness. The following proportions are in accordance with good practice:

b, breadth of cotter $= 1.3d_1$;

t, thickness of cotter $= .3d_1$;

d, diameter of pierced rod $= 1.2d_1$;

D, diameter of socket in front of cotter $= 2.4d_1$ or $2d$.

D_1, diameter of socket behind cotter $= 2d_1$;

D_2, diameter of collar on rod $R = 1.5d_1$;

t, thickness of collar on rod $R = \frac{1}{2}d_1$;

l, the length of the rod and socket beyond the cotter = from $\frac{3}{4}d$ to d.

When d is known the diameter of the solid rod $(d_1) = .82d$. The clearance c may be made $\frac{1}{8}''$. The cotter need not extend beyond the greatest diameter of the socket more than $\frac{1}{8}''$ when driven home.

Fig. 84 shows an arrangement often used for securing an engine piston-rod to the piston. Here, instead of having a collar on the rod R to resist the thrust, the rod-end is tapered.

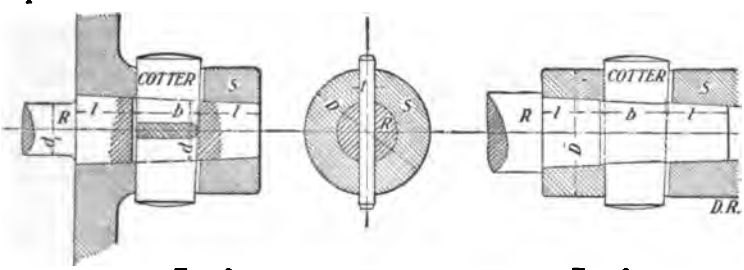


FIG. 84. FIG. 85.

In Fig. 85, the pierced part of the rod has a smaller diameter than the solid rod. Such a condition is possible when the diameter of the rod is increased in consequence of its having to resist buckling stresses. The joint being subjected only to tension and compression, the rod would under these conditions be excessively strong if proportioned to the diameter of the solid rod. We must therefore find the diameter (d_1) and proportion the joint independently of the actual rod diameter. d_1 is found by the formula—

$$\left.\begin{aligned} &\frac{\pi d_1^2}{4} f_t = P \\ \text{from which} \qquad &d_1 = \sqrt{\frac{P}{.7854 f_t}} \end{aligned}\right\} \quad \ldots \ldots \quad (19)$$

Where P is the pull on the rod. For steel rods f_t may be taken at 7000 and 5000 for wrought iron. The taper of the

rod-end may be made from $\frac{1}{2}''$ to $1''$ per foot of length, i.e., from 1 in 12 to 1 in 24. The diameter d on the tapered rod-end is taken, when the cotter-way is curved at the end, where the curve begins, as in Fig. 84, and at the end of the cotter-way when the cotter-way is rectangular.

Exercise 25.—Draw a SECTIONAL ELEVATION, a HALF PLAN, and HALF SECTIONAL PLAN of the cotter-joint shown in Fig. 83. Make $d = 2''$. *Scale full size.*

Exercise 26.—Design cotter-joints suitable for fastening a steel piston-rod to the piston and cross-head, as shown in Figs. 84 and 85. Make the diameter of the rod $d_s = 2\frac{3}{8}''$ and assume that the rod is subjected to a load of 9000 lbs. The rod-ends having a taper of 1 in 12. *Scale full size.*

Construction.—Having determined the diameter (d) of a rod suitable for resisting tensile stresses, then from d find the other proportions of the joints, as in Exercise 25. Measure off the distances l and b along the centre line and mark off the diameter (d) at the proper point according to the shape of the cotter in cross-section, then in the manner given in the construction in connection with Exercise 13 draw the rod-end to the given taper. The construction for finding the taper need not be inked in. Complete the drawing, filling in the actual dimensions and leaving off all reference letters.

COTTER AND GIB.

When one of the pieces connected by the cotter is a thin strap, as in Fig. 86, a second cotter, called a gib, is used. The gib is provided with a head at the ends which project over the strap S, thus preventing it

(the strap) from being forced open by the friction between it and the cotter as the latter is driven into place. Figs. 86 and 89 show the application of gib and cotter to strap-end connecting-rods, where R is the rod and S the strap. When two gibs are used, as in Fig. 88, the sliding surface on each side of the cotter is the same. Instead of having both gibs tapered, as shown in Fig. 88, one of them may be parallel and the taper all on one side of the cotter. The strength of the gib and cotter in combination is made the same as the

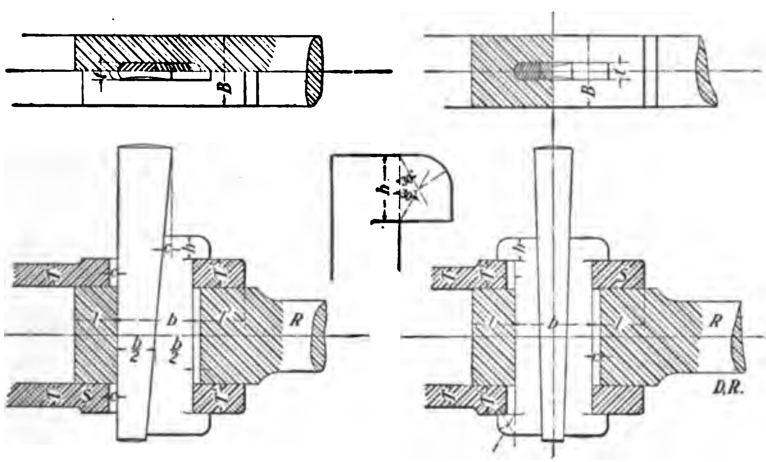


FIG. 86. FIG. 87. FIG. 88.

single cotter and should be proportional to the strap S. The working strength of the strap at the thinnest part is found by the equation

$$\left.\begin{aligned} 2BTf_t &= P. \\ \text{from which} \qquad T &= \frac{P}{2Bf_t} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

where P is the maximum pull on the rod, T the thickness,

and B the breadth of the strap. Then as the gib and cotter are to have the same strength as the single cotter, and as B is equal to, or a little greater than d (the diameter of the rod), t may be made equal to $.25B$ and

$$b = 1.3\sqrt{\frac{2BT}{.7854}},$$

T', the thickness of the strap where it is pierced by the cotter, should not be less than $1.3T$. l', the distance from the gib to the end of the strap, $= 2T$. l, the distance from the cotter to the end of the rod, $= 1.5T$. c, the clearance, should not be less than c' (the difference between the widest part of the cotter and the width of the cotter at the top of the gib-head). The method of constructing gib-heads is shown in Fig. 87, where h, the height of the gib-head, $= 1\frac{1}{4}t$.

Cotter-locking Arrangements.—A simple method, and one that is used in nearly all cases where it is possible, is to screw one or two set-screws through the rod until the point or points press against the cotter. To keep the burs, raised by the point of the screw, from interfering with the motion of the cotter, the set-screw bears on the bottom of a shallow groove cut on the side of the cotter, as shown in Fig. 89. The diameter of the set-screw need not exceed $\frac{5}{8}''$. The length of the groove is equal to the travel of the cotter + the diameter of the set-screw. The travel of the cotter is the distance from the top of the gib (or where no gib is used, from the top of the piece into which the cotter passes) to the top of the cotter when the cotter is just in place.

The width of the groove is equal to the diameter of the set-screw point, and the depth $= \frac{1}{16}''$.

In Fig. 90 the cotter is locked by an upper and lower nut upon a screwed extension of the gib, which passes through a head formed on the cotter. This arrangement is used for fastening in, and may be used for forcing the cotter into, position.

d, the diameter of the screw $= t$;
h, the height of the head $= 1\frac{1}{4}d$.

As the axis of the locking-screw is not parallel with the side of the cotter that is in contact with the gib, the hole in the cotter head through which the screw passes is elongated

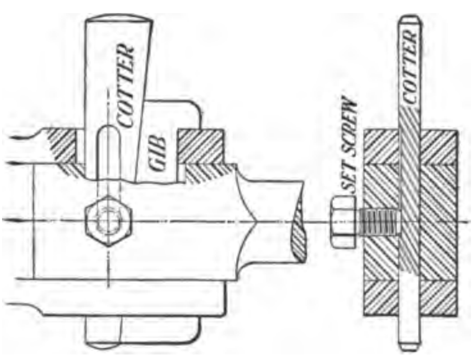


FIG. 89.

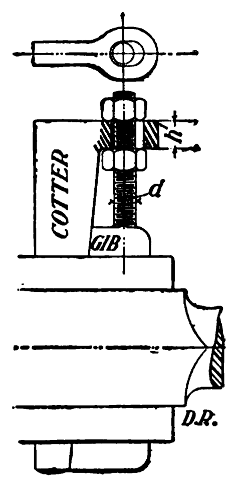


FIG. 90.

to an amount equal to the taper of the cotter in its length of travel $+ \frac{1}{16}''$ for clearance.

Exercise 27.—Draw a SECTIONAL ELEVATION and a HALF SECTIONAL PLAN, a PLAN, and SECTIONAL END VIEW of a gib- and cotter-joint to resist a tension of 12,000 lbs. Make the diameter (d) of the rod $= 2''$. The cotter to have an adjust-

ment of $\frac{3}{8}''$ with a taper = 1 in 12. Show the method of locking the cotter by means of a set-screw. *Scale full size.*

Exercise 28.—Draw a SECTIONAL ELEVATION AND PLAN of a double gib- and cotter-joint for the same conditions as given in Exercise 27. Show the method of locking the cotter by means of an upper and lower nut upon a screwed extension of the gib. *Scale full size.*

CHAPTER III.

RIVETS AND RIVETED JOINTS.

RIVETS are made from round bars of steel, wrought iron, copper, or brass, and are used to fasten two or more plates permanently together.

The plates to be riveted are either drilled or punched with holes $\frac{1}{16}''$ larger in diameter than that of the rivet-shank.

When the rivet is placed in position through the plates a sufficient length of shank projects beyond the plates to provide for forming the *rivet-point head* either by hammering or by machine-pressure (see Fig. 92). Unwin calls a riveted joint the "simplest permanent fastening."

Rivets are made by being pressed into shape while red-hot with rivet-making machines using dies of suitable size and form.

The names and proportions of rivet-heads shown by Figs. 92 to 96 will be given later.

The end of the rivet opposite to the head before riveting up is called the point and after riveting the point-head.

Just before using the rivet is heated red-hot and when placed in position for hand-riveting is held there by means of a large hammer with a long handle fulcrumed at a convenient distance from the rivet and a man's weight applied at the end, while the point-head is made by two riveters either in the form of the steeple head by hammers only or the snap head (Fig. 92) by using a cup-shaped die called a snap.

In machine-riveting the point-head is pressed into shape by suitable dies, the motive power being either a lever, steam, hydraulic, or pneumatic pressure. Machine-riveting upsets the rivet and fills the hole much better than hand-riveting, because the steady even pressure of the former is exerted uniformly through the whole of the rivet.

Hydraulic riveting is preferred to steam-riveting, because the pressure from the former can be gradually applied, while the force from the latter generally comes upon the rivet with such rapid blows that sufficient time is not allowed for the rivet to properly fill the hole.

Rivet-holes punched through rigid steel plates should always be annealed after punching, because the punching injures the material surrounding the hole to such a dangerous extent that the elasticity of the plate is destroyed, and when the joint is subjected to strain the stress is not uniformly distributed between the rivet-holes. Another difficulty with punched holes is the imperfect spacing of the rivet-holes.

Drilled holes are usually more expensive than punched holes and the sharp square edge is not as favorable to the resistance of the rivet to shearing, but they are more accurate in size and spacing, and the resistance of the rivet to shearing can be increased by slightly rounding the edges of the holes.

Calking.—No riveted joint is ever perfectly steam-tight without calking. This is a process by which a narrow strip of the bevelled edge of one plate is brought into forcible contact with the plate beneath it.

At *a*, Fig. 91, is shown the calking-tool commonly used in hand-calking, and at *b* an improved form of calking-tool patented bv Mr. J. W. Connery of Philadelphia and known as

the concave calking-tool from the concave finish given to the calked edge of the plate. This is a favorite style of calking with locomotive-builders for high-pressure boilers.

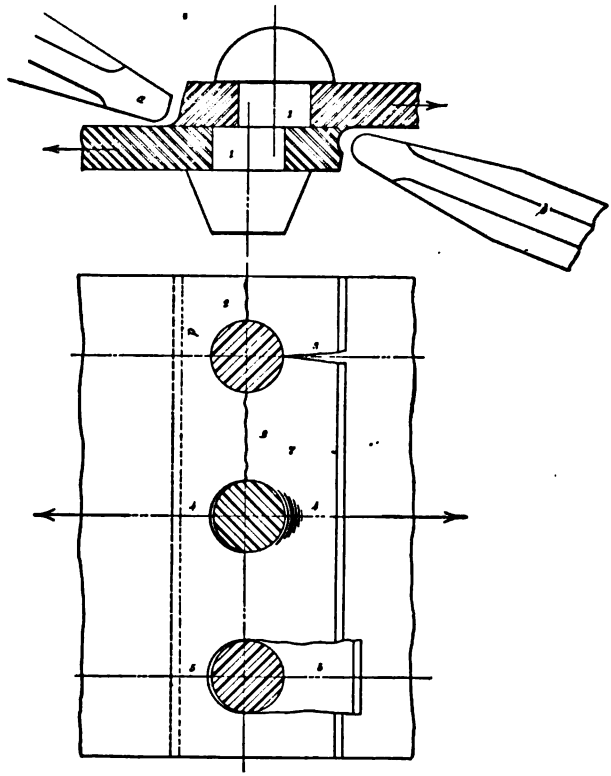


FIG. 91.

Calking with pneumatic calking-hammers has become quite general in most first-class boiler-shops. Peabody and Miller in their "Steam-boilers" describe a pneumatic calking-machine as follows:

"In general principle it resembles a rock-drill and consists of a cylinder in which works a piston and rod on the end of

which is the calking-tool. Air is supplied for working the piston at a pressure of 50 or 60 lbs. through a flexible tube. It makes about 1500 working strokes a minute $\frac{3}{16}''$ long. The calker which is about $2\frac{1}{2}''$ in diameter outside and $15''$ long over all, is held by a workman who presses it slowly along the seam to be calked. The edge of the tool is well rounded, so as not to injure the lower plate. Work can be done four times as rapidly with the pneumatic calker as by hand."

The edges of rivet-heads are not calked except when they show a leak during the process of testing. In some of the largest boiler-shops an inspector is employed part of whose duty it is when examining a boiler to discover if any of the rivets are loose. This is done by placing a finger on the under side of the suspected rivet and tapping the top of it with a small hammer made for the purpose; if the rivet is not perfectly tight it will be easily detected by the finger; in such a case the loose rivet is cut out and replaced by a new one.

The Forms of Rivets.—The standard forms of rivets in general use are: (1) the *button* head (Fig. 92); (2) the

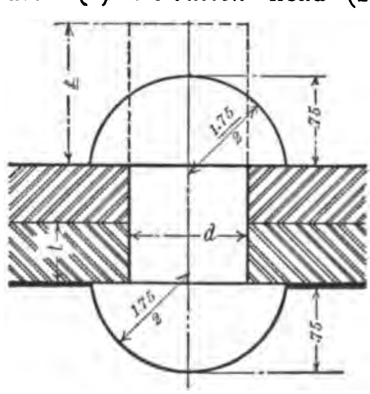


FIG. 92.

conical head (Fig. 93); (3) the *steeple* head (*b*) (Fig. 94); (4) the *steeple* head (*d*) (Fig. 95); (5) the *countersunk* head (Fig. 96).

The *button* head, or, as it is sometimes named, the *snap* head, is usually made with a machine-riveter.

The *conical* is also a machine-formed head and is commonly used with a button point-head or tail and sometimes with a steeple point.

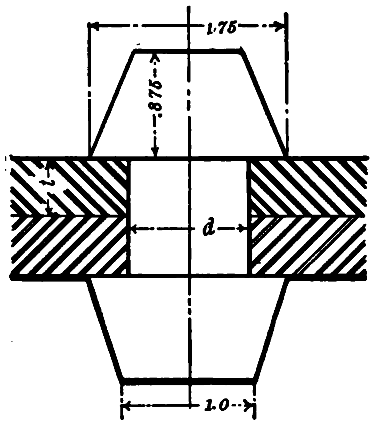


Fig. 93.

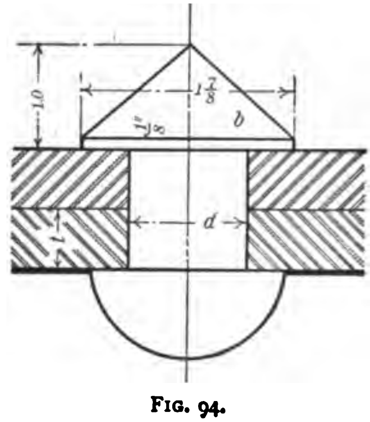


Fig. 94.

The *steeple* point-head is the form mostly used in hand-riveting.

The *countersunk* point-head is only used when there is not sufficient room for one of the other forms and should never be used unless it is impossible to avoid it. It is more costly than, and not as strong as, the other forms.

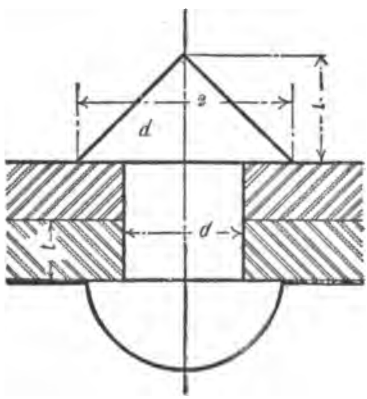


FIG. 95.

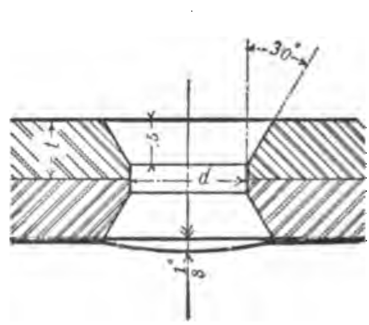


FIG. 96.

Proportions of Rivet-heads.—The proportions given in the figures in terms of the diameter *d* are those used by the Champion Rivet Co. and agree closely with general practice.

Length of Rivet-shank.

The length *L* (Fig. 92) for countersunk point-head and 2 plates	$1d$
For countersunk point-head and 3 plates	$1d+\frac{1}{8}''$
For steeple point-head	$1.25d$
For steeple point-head, large, machine-driven	$1.50d$
For button point-head	$1.25d$

The above proportions are good for ordinary boiler-plates, but, since the holes are $\frac{1}{16}''$ larger than the rivet, the shank

should be increased in length for *thick* plates to properly fill the additional annular space.

The rivet-shank is usually about $\frac{1}{16}''$ smaller in diameter than the hole and has a slight taper toward the point.

Exercise 29.—Make a drawing of each style of riveting shown in Figs. 92 to 96, making t equal to $\frac{3}{4}''$ and selecting from Table 14, page 135, the diameter of rivet. For conventions see page 22. *Scale full size.*

Riveted Joints.—There are in common use at least five different styles of riveted joints, viz.: the single-riveted lap-joint (Fig. 97); the double-riveted lap-joint with *staggered* spacing (Fig. 98); the double-riveted lap-joint with *chain* spacing (Fig. 99); the single-riveted butt-joint with *chain* spacing (Fig. 100); the double-riveted butt-joint; the multiple-riveted lap-joint which has more than two rows of rivets in the lap; the multiple-riveted butt-joint which has more than two rows of rivets on each side of the line where the plates butt together (Fig. 103).

NOTATION.

d = the diameter of the rivet-hole or of rivet when riveted up.

p = the pitch of the rivets, i.e., the distance from the centre of one rivet to the centre of the next in the same row (Fig. 97).

l = the distance from the centre of rivet-hole to edge of plate (Fig. 97).

r = the distance between the rows on double-riveted joints.

r_1 = the distance between outside rows of rivets on lap-joints with welt-strip and butt-joints.

m = the least distance between the edge of rivet-hole and edge of plate = *margin* (Figs. 97 to 103).

t = the thickness of plate.

t_1 = thickness of outside welt-strips for butt-joint.

t_2 = thickness of inside welt-strips for butt-joint.

t' = thickness of inside welt-strips for lap-joint.

f_t = the tensile strength per square inch of the plate in lbs.

f_s = the shearing strength per square inch of the rivet in lbs.

f_s' = the shearing strength per square inch of the plate in lbs.

f_c = the compressive or crushing strength per square inch of the plate in lbs.

R = the radius of boiler in inches on the outside of course of smallest diameter.

N = the width of widest welt-strip.

K = the width of narrowest welt-strip.

P = working pressure in lbs. per square inch.

D = outside diameter of boiler-shell at course of smallest diameter.

F = factor of safety.

E = efficiency of riveted joint.

T = total tensile stress.

a = area of rivet-hole = $.7854d^2$.

Strength of Single-riveted Joint.—There are five different ways in which a single-riveted lap-joint may give way:

(1) Shearing the rivet, as shown at 1 in Fig. 91.

(2) Tearing plate along the centre line of rivets, shown at 2, 2.

(3) Tearing the plate through the margin, shown at 3.

(4) Crushing the rivet or the plate in front of the rivet (4, 4).

(5) Shearing the plate in front of the rivet (5, 5).

The shearing strength of the rivet

$$= \frac{\pi d^2}{4} \times f_s = a \times 38,000. \quad . \quad . \quad . \qquad (1)$$

The resistance of plate to tearing on centre line of rivet

$$= (p - d)t \times f_t. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

The resistance of the plate to tearing at 3 has been found by experiment to be great enough when the distance l is made equal to $1\frac{1}{2}d$, and, as this rule agrees with general practice, it will be maintained throughout this work.

The compressive resistance of the plate at 4 is

$$t \times d \times f_c. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

The resistance to shearing the plate in front of the rivet as shown at 5, 5,

$$= 2t \times l \times f_s'. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

But if the joint is made strong enough to resist shearing the rivet or tearing the margin it will be strong enough to resist shearing or crushing the plate in front of the rivet, so that the latter may generally be disregarded.

The thickness of the boiler-plate is

$$t = \frac{P \times R \times F}{f_t \times E} = \frac{PR}{11,000E}. \quad . \quad . \quad . \qquad (5)$$

The value for t should be taken as the nearest even sixteenths of an inch. Take $E = .70$.

The thickness of dome-sheet may be calculated by the same formula.

In locomotive-boilers the thickness of tube-sheets for $\frac{3}{8}''$ shells and over should be $\frac{1}{2}''$ to $\frac{9}{16}''$.

When shells are less than $\frac{3}{8}''$ thick it is usual to make the thickness of tube-sheets equal to $t + \frac{1}{8}''$.

The throat-sheet is usually made $\frac{1}{8}''$ thicker than the shell to allow for extra flanging.

In thick shells, $\frac{3}{4}''$ or over, $\frac{1}{16}''$ thicker will be sufficient.

When the back tube-sheet is separated from the fire-box throat-sheet the latter should be made the same thickness as the fire-box side sheets, viz., $\frac{5}{16}''$.

The fire-box crown-sheet is usually made $\frac{3}{8}''$ and the side and door sheets $\frac{5}{16}''$ thick.

Diameter d of Rivet-hole.—It is very desirable in designing riveted joints to obtain the highest efficiency and still maintain a proper tightness by using a pitch not too long for calking.

In determining the diameter d of the rivet it is necessary that it should be strong enough to resist both shearing and crushing. Now the resistance to shearing is

$$\frac{\pi d^2}{4} f_s ,$$

while that of crushing is

$$d t f_c ,$$

which shows that the latter increases as the diameter and the former as the square of the diameter. So that if we can obtain such a relation between the length of the pitch and the

diameter of the rivet-hole as will give the highest efficiency consistent with tightness the crushing strength of the rivet or the plate in front of the rivet need not be considered.

To our knowledge the maximum limit for the length of pitch that will insure perfect tightness of the joint has never been ascertained by experiment or test, so that we have to depend largely on existing practice in determining the ratio between *d* and *t*.

Mr. Wm. M. Barr in his "Boilers and Furnaces" gives the following ratios between the thickness of the plate and the diameter of the rivet for single-riveted lap-joints, using the nearest even sixteenths of an inch, for steel plates and steel rivets (tensile strength of plates 55,000 lbs. and shearing strength of rivets 44,625 lbs. per square inch):

TABLE 14.

t	Ratio of *d* to *t*.	*d*	Decimal Equivalent.	Area of *d*. Sq. In.	Pitch of Rivets.	
					Decimal.	Working Fraction.
$\frac{1}{4}''$	2.75	$\frac{11}{16}''$	.6875	.371	1.892	$1\frac{7}{8}''$
$\frac{5}{16}''$	2.40	$\frac{3}{4}''$	.75	.442	1.897	$1\frac{7}{8}''$
$\frac{3}{8}''$	2.17	$\frac{13}{16}''$	.8125	.518	1.934	$1\frac{15}{16}''$
$\frac{7}{16}''$	2.00	$\frac{7}{8}''$	.875	.601	1.990	$2''$
$\frac{1}{2}''$	1.87	$\frac{15}{16}''$	.9375	.690	2.058	$2\frac{1}{16}''$
$\frac{9}{16}''$	1.78	$1''$	1.000	.7854	2.133	$2\frac{1}{8}''$
$\frac{5}{8}''$	1.70	$1\frac{1}{16}''$	1.0625	.887	2.205	$2\frac{3}{16}''$
$\frac{11}{16}''$	1.64	$1\frac{1}{8}''$	1.125	.994	2.298	$2\frac{5}{16}''$
$\frac{3}{4}''$	1.58	$1\frac{3}{16}''$	1.1875	1.108	2.386	$2\frac{3}{8}''$

A committee of the Railway Master Mechanics' Association on riveted joints in 1895 gave the following ratios between *d* and *t* in their report for single-riveted lap-joints (steel plates of 55,000 lbs. tensile strength and iron rivets of 38,000 lbs. shearing strength per square inch)

TABLE 15.

t	Ratios.	Mean Ratios.	d	p	E
$\frac{1}{4}''$	2.25 to 3.00	2.62	$\frac{11}{16}''$	$1\frac{3}{4}''$	58.6%
$\frac{5}{16}''$	2.00 to 2.80	2.40	$\frac{3}{4}''$	$1\frac{3}{4}''$	55.8%
$\frac{3}{8}''$	2.00 to 2.60	2.30	$\frac{7}{8}''$	$2''$	55.3%
$\frac{7}{16}''$	1.71 to 2.42	2.06	$\frac{15}{16}''$	$2\frac{1}{16}''$	52.8%
$\frac{1}{2}''$	1.75 to 2.35	2.05	$1''$	$2\frac{1}{16}''$	51.5%
$\frac{9}{16}''$	1.77 to 2.33	2.05	$1\frac{3}{16}''$	$2\frac{9}{16}''$	53.0%
$\frac{5}{8}''$	1.60 to 2.10	1.85	$1\frac{3}{16}''$	$2\frac{7}{16}''$	50.2%

Pitch p of Rivets.—The total strength of a boiler-plate is reduced by the rivet-holes, and the shorter the pitch the weaker the plate, but on the other hand if the pitch is too long the rivet will shear unless it is increased in diameter to correspond in shearing strength to the tensile strength of the net section of plate, but a long pitch and large rivet diameter are also limited by the fact that under high pressures such a joint is hard to make tight.

The mean ratios between the thickness t of plate and the diameter d of the hole given in Table 15 are recommended as good modern practice.

To find the pitch p in terms of the thickness of the plate t and the diameter d:

$$p = \frac{\pi d^2 f_s}{4 t f_t} + d. \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Exercise 30.—Design a single-riveted lap-joint for a boiler 48″ diameter and carrying a steam-pressure of 148 lbs. per square inch, plates to be soft steel of 55,000 lbs. tensile strength per square inch and iron rivets of a shearing strength = 38,000 lbs. per square inch. *Scale 6″ = 1 foot.*

(1) Find thickness t of plate by formula 5, page 132.

(2) Determine diameter d of rivet from the mean ratio in Table 2.

(3) Calculate the pitch p by formula 6, page 138.

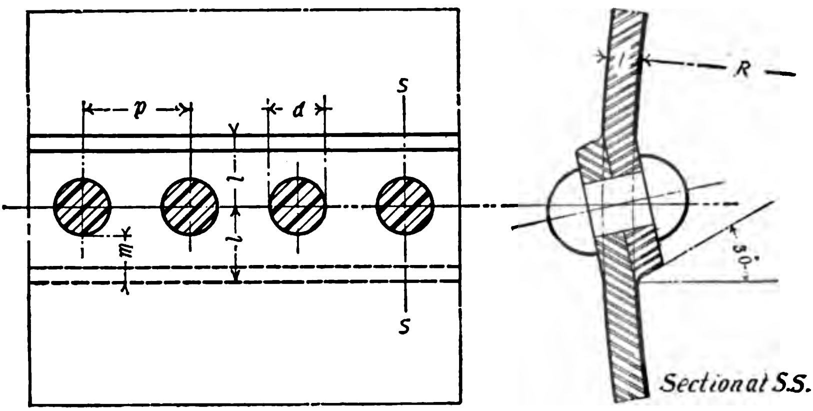


FIG. 97.

Make complete drawings as shown in Fig. 97, giving actual dimensions in place of letters.

Single-riveted lap-joints are commonly used for circumferential seams of steam-boilers.

To determine whether a circumferential seam should be single- or double-riveted let us take the following example:

Diameter of boiler 48″.

Steam-pressure per square inch 148 lbs.

Diameter of rivet = .875″.

Pitch = 2″.

Thickness of plate = .375″.

The total force will be

$$.7854 D^2 P = 1809.6 \times 148 = 267{,}820.8 \text{ lbs.} \quad . \quad (7)$$

The resistance due to the rivets

$$= \frac{.7854 d^2 \times n \times f_s}{F} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

n = the number of rivets in the circumferential seam.

F = the factor of safety = 6.

Therefore, substituting, we have

$$\frac{.601 \times 75 \times 38,000}{6} = 285,475 \text{ lbs.},$$

and, subtracting the force from the resistance, we have a difference of 17,654.2 lbs. in favor of the rivets.

The total resistance of the plate is

$$\frac{(p-d) \times t \times f_t \times n}{F} = \frac{1.125 \times .375 \times 55,000 \times 75}{6} \qquad (9)$$

$$= 288,750 \text{ lbs.},$$

and, subtracting the total force, 267,820 lbs., from 288,750, there remains a difference of 20,929 lbs. in favor of the plate, which shows that a single-riveted lap-joint is strong enough for the circumferential seams of a boiler of the above dimensions.

Prof. Lanza referring to the efficiency of riveted joints in his "Applied Mechanics" says:

"A riveted joint of maximum efficiency should fracture the plate along the line of rivets, for it is clear that if failure occurs in any other manner, as by shearing the rivets or tearing out the rivet-holes, there remains an excess of strength along the line of riveting, or, in other words, along the net section of plate—if in a single-riveted joint—which has not been made use of; but when fracture occurs along the net section an excess of strength in other directions is immaterial.

"If the strength per unit of metal of the net section is constant it would be a very simple matter to compute the efficiency of any joint, as it would be merely the ratio of the net to the gross areas of the plate.

"The tenacity of the net section, however, varies and this variation extends over wide limits."

This being so, the pitch in the last example is slightly longer than is necessary.

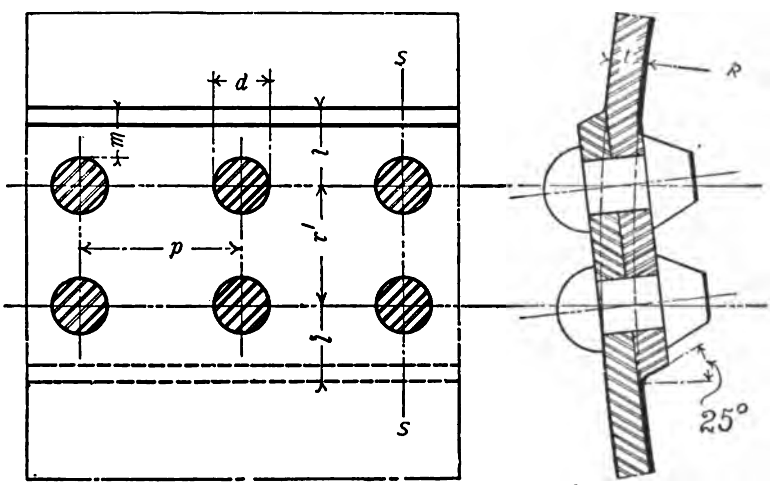
Double-riveted Lap-joints.—The arrangement of the rivets in Fig. 98 is called *chain* riveting and in Fig. 99 *zigzag* riveting. The double-riveted joint is stronger than the single-riveted joint because of the greater net section of *plate* and smaller diameter of rivet-holes. All longitudinal seams in steam-boilers should be at least double-riveted. Steel plates and *iron rivets* are considered the safer practice because of the danger of overheating the *steel* rivets.

Wm. M. Barr in his "Boilers and Furnaces" referring to the heating of steel rivets says: "It is important that steel rivets be uniformly heated throughout, and not the points merely, as is the ordinary method of heating iron rivets; neither should they be heated as highly as iron rivets, and should never exceed a bright cherry-red. Particular attention should be given to the thickness of the fire.

"If excluded from free oxygen steel cannot be burned; if the temperature is high enough it can be melted; but burning is impossible in a *thick* fire with moderate draft."

Chain riveting with rivets of the same pitch has been found by experiment to be stronger than the zigzag riveting. See Barr's "Boilers and Furnaces," page 85, where it states that the lap is wider for chain riveting, "and no doubt the fric-

tion of this wider joint contributes towards the observed increase in strength," but the late D. L. Barnes and others who have tested riveted joints state that the friction between the plates cannot be considered, because long before the ultimate strength of the lap is reached the plates are so far apart that "you can stick a knife-blade between them." The zigzag riveting is preferred in locomotive-boiler seams, because the joints are tighter under the high pressures carried than they would be with the wider lap of the chain riveting.



Section at SS.

FIG. 98.

Exercise 31.—Make the drawings for a double-riveted lap-joint, chain riveting, like Fig. 98, except that the actual dimensions should be given instead of the letters shown. Steel plates and iron rivets. Thickness of plate $= \frac{5}{8}''$, $p = 3\frac{5}{16}''$, $d = 1\frac{1}{8}''$, $l = 1\frac{1}{2}d$, $r' = 2d + \frac{1}{4}''$, $R = 30''$. *Scale 6" = 1 ft.* Calking need not be shown now.

Calculate the efficiency of this joint in comparison with he strength of the plate.

Taking f_t at 55,000 and f_s at 38,000 as before, the total strength of solid plate is

$$p \times t \times f_t = 3.3125 \times .625 \times 55{,}000 = 110{,}000 \text{ lbs.}$$

The strength of the net section of plate is

$$(p - d)tf_t = (3.3125 - 1.125).625 \times 55{,}000 = 75{,}735.$$

The shearing strength of the rivets $= .7854d^2 \times 38{,}000 \times 2$ (for 2 rivets) $= 75{,}544$, nearly equal to the strength of the net section of the plate. Therefore the efficiency of the joint is equal to

$$E = \frac{75{,}544}{110{,}000} = 69 \text{ per cent nearly.}$$

The following ratios of d to t for double-riveted joints were calculated from the report of a committee on riveted joints to the Am. Ry. M. M. Association in 1895:

TABLE 16.

t	Ratios, Max. and Min.	Ratios, Mean.	d	p	a Area of Rivet.	E
$\frac{3}{8}''$ (.375)	2.00 to 2.66	2.33	$\frac{7}{8}''$	$3\frac{1}{16}''$	.6	71.4
$\frac{7}{16}''$ (.4375)	1.71 to 2.42	2.06	$\frac{15}{16}''$	$3\frac{1}{8}''$	.69	69.7
$\frac{1}{2}''$ (.5)	1.75 to 2.375	2.063	$1\frac{1}{16}''$	$3\frac{1}{4}''$	.8866	69.6
$\frac{9}{16}''$ (.5625)	1.77 to 2.22	1.99	$1\frac{1}{8}''$	$3\frac{9}{16}''$	.994	68.4
$\frac{5}{8}''$ (.625)	1.60 to 2.00	1.80	$1\frac{1}{8}''$	$3\frac{5}{16}''$	.994	66 0
$\frac{11}{16}''$ (.6875)	1.54 to 1.909	1.72	$1\frac{3}{16}''$	$3\frac{7}{16}''$	1.107	64.7
$\frac{3}{4}''$ (.75)	1.416 to 1.75	1.58	$1\frac{3}{16}''$	$3\frac{1}{4}''$	1.107	62.7

To find the *pitch* p for double-riveted lap-joints with steel plates and iron rivets.

$$p = \frac{2 \times a \times f_s}{tf_t} + d = \frac{2 \times a \times 38{,}000}{t \times 55{,}000} + d. \quad . \quad (10)$$

To find the distance between the centres of rows of rivets r (Fig. 99).

Prof. Kennedy gives $\frac{2p + d}{3}$ for the diagonal pitch. r may be found graphically or calculated by formula

$$r = \frac{\sqrt{(7p + 2d)(p + 2d)}}{6} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Table 17 gives the distances (r) calculated by this formula for the different sizes of rivets.

Exercise 32.—Make drawings as per Fig. 99 of a double-riveted lap-joint, zigzag riveting. $t = \frac{5}{8}''$, ratio of d to $t =$

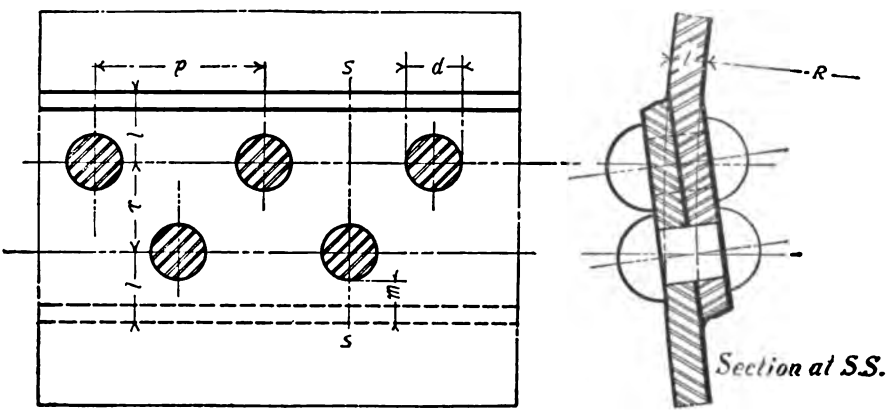


FIG. 99.

1.80, $R = 30''$. $l = 1\frac{1}{2}d$ in even $\frac{1}{16}''$. *Scale 6'' = 1 foot.*

Find r by formula 11. Find p by formula 10.

Exercise 33.—Make drawings similar to those in Fig. 100 showing the junction of a double zigzag-riveted longitudinal seam with a single-riveted circumferential seam for a steam-boiler. $t = \frac{9}{16}''$, d calculated from the mean ratio in Table 16, p to be determined from formula 10, p' from Table 14, $R = 29''$, r may be calculated from formula 11. *Scale 6'' = 1 foot.* Actual dimensions to be placed on drawing where letters

show in figure. Steel plates and iron rivets. Finish sheet according to directions given on pages 19 and 20.

Lap-joints with Inside Welt-strip.—This style of riveting, shown in Fig. 101, is used for both single- and double-

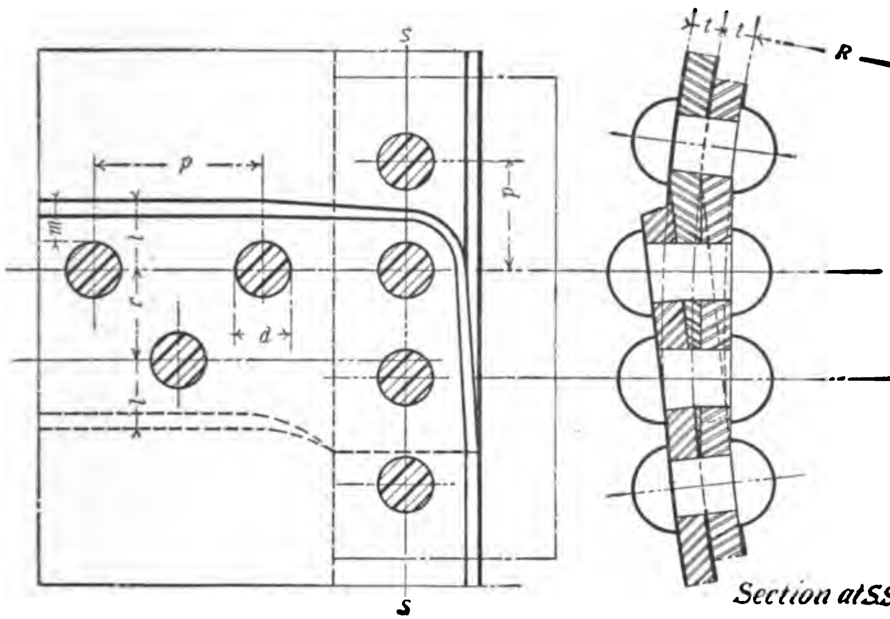


FIG. 100.

riveting and possesses some of the features of the butt- and lap-joint. In the single-riveted joint of this kind the middle row of rivets which rivet the three thicknesses of plate should be spaced according to the rule given for p in the single-riveted lap-joints on page and the spacing of the outer rows $= 2p$.

These joints are better than the simple lap-joint, but are more expensive, and are not any better than the butt-joint (Fig. 102), which is simpler and less expensive.

The double-riveted lap-joint with inside welt (Fig. 101) may fail in any one of the following ways:

(1) By shearing the rivets holding plate (a).

Resistance against shearing $= 5a \times f_s = 5a \times 38,000$. (12)

(2) By tearing plate (a) along the outside row of rivets.

Resistance against tearing plate as above

$$= (2p - a)t \times f_t = (2p - a)t \times 55,000. \quad . \quad . \quad (13)$$

(3) By tearing plate (a) along the intermediate row + the shearing of one rivet.

$$\text{Resistance} = (2p - 2a)t \times 55,000. \quad . \quad . \quad (14)$$

$$\text{Strength of solid plate} = 2p \times t \times f_t. \quad . \quad . \quad (16)$$

$$E = \frac{\text{least resistance}}{\text{strength of solid plate}}. \quad . \quad . \quad . \quad . \quad (17)$$

Exercise 34.—Make complete drawings of a double-riveted lap-joint with inside welt, zigzag spacing, Fig. 101. The sectional view of this figure is wrongly projected with intention. Student must make correct projection.

Take the remaining dimensions from the following table:

TABLE 17.

DOUBLE-RIVETED LAP-JOINTS WITH INSIDE WELTS.

t	d	p	m	r	r_1	N	Efficiency.
$\frac{3}{8}''$	$\frac{15}{16}''$	$3\frac{5}{8}''$	$\frac{15}{16}''$	$2\frac{1}{16}''$	$3\frac{9}{16}''$	$12''$	87.0
$\frac{7}{16}''$	$1''$	$3\frac{5}{8}''$	$1''$	$2\frac{1}{16}''$	$3\frac{13}{16}''$	$12\frac{11}{16}''$	85.5
$\frac{1}{2}''$	$1\frac{1}{8}''$	$4''$	$1\frac{1}{8}''$	$2\frac{5}{16}''$	$4\frac{1}{8}''$	$13\frac{15}{16}''$	85.8
$\frac{9}{16}''$	$1\frac{1}{16}''$	$4''$	$1\frac{3}{16}''$	$2\frac{5}{16}''$	$4\frac{3}{8}''$	$14\frac{5}{8}''$	85.0
$\frac{5}{8}''$	$1\frac{1}{4}''$	$4''$	$1\frac{1}{4}''$	$2\frac{3}{8}''$	$4\frac{1}{2}''$	$15''$	84.3

The Double-riveted Butt-joint with Inside and Outside Welts.—This style of joint is a very common one for longi-

tudinal seams of steam-boilers with plates $\frac{3}{8}''$ thick and over. As shown by Fig. 102, the boiler-shell is rolled to a perfect cylinder and the two edges of the plate which butt together

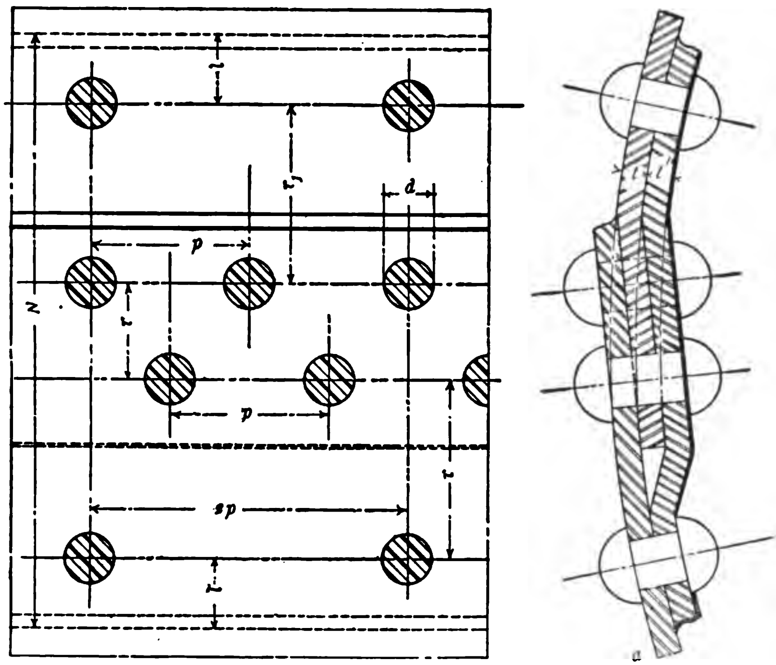


FIG. 101.

are held by two welt-strips riveted to each other and to the ends of the plate.

In a repeating section of the plate $= 2p$ there are two rivets in double shear and two half rivets in single shear.

From experiments made by the English Admiralty and others it has been demonstrated that 1 rivet in double shear is equal to 2 rivets in single shear. For convenience we will assume this to be so at present, although it is quite usual for designers of steam-boilers to use a value of from 1.75 to 1.90

for rivets in double shear; and, as the latter values agree more nearly with general practice for butt-joints, it will be necessary for us to modify our proportions in this regard, as will appear later. Therefore to prevent the plate a pulling out from between the welt-strips the resistance to shearing will be

$$5 \times a \times f_s,$$

there being two rivets in double shear and two half rivets in single shear = 5 areas in single shear.

Resistance to tearing the net section of plate at the outer row is

$$(2p - d)tf_t.$$

Resistance to tearing the plate between the inner row of rivets and shearing rivets in outer row is

$$(2p - 2d)t \times f_t + 1af_s.$$

Resistance to crushing the plate in front of 3 rivets is

$$3tdf_c.$$

f_c may be taken at 80,000 lbs. per square inch for iron and 90,000 for steel rivets.

Strength of whole plate equal in width to $2p$ is

$$2p \times t \times f_t.$$

Exercise 35.—Draw *elevation* and *cross-section* of a double-riveted butt-joint with outer and inner welts similar to Fig. 102, given $t = \frac{9}{16}''$, $d = 1.92t$.

If we consider the resistance to tearing equal to the resistance to shearing, then

$$2p = \frac{5 \times a \times f_s}{tf_t} + d, \quad . \quad . \quad . \quad . \quad (18)$$

but this makes the pitch too long, because of the excess of strength in the rivets against shearing. A better proportion

and one that conforms to good practice is

$$2p = \frac{.85(5af_s)}{tf} + d.$$

t_1 and t_2 are usually equal to t, but occasionally t_1 will be

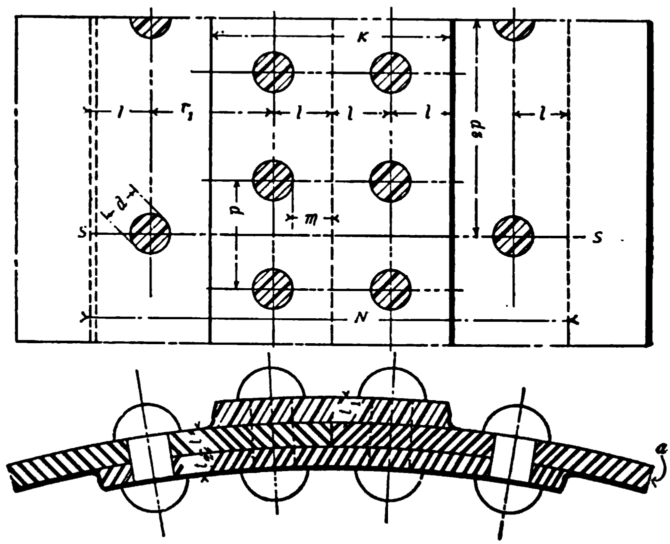


FIG. 102.

found $\frac{1}{16}''$ thicker than t. The Hartford Steam-boiler Inspection & Insurance Company give all welt-strips $\frac{1}{16}''$ less in thickness than t.

For the remaining dimensions see the following table:

TABLE 18.

For double-riveted butt-joints with outer and inner welt-strips.

t	Ratio of d to t. Average.	Diameter of Hole. d	Pitch. p	l	$r_1 = 2l$	K	N
$\frac{3}{8}''$	2.19	$\frac{13}{16}''$	$2\frac{7}{16}''$	$1\frac{3}{16}''$	$2\frac{3}{8}''$	$4\frac{3}{4}''$	$9\frac{1}{2}''$
$\frac{7}{16}''$	1.93	$\frac{7}{8}''$	$2\frac{15}{32}''$	$1\frac{5}{16}''$	$2\frac{5}{8}''$	$5\frac{1}{4}''$	$10\frac{1}{2}''$
$\frac{1}{2}''$	1.92	$\frac{15}{16}''$	$2\frac{1}{2}''$	$1\frac{3}{8}''$	$2\frac{3}{4}''$	$5\frac{1}{2}''$	$11''$
$\frac{9}{16}''$	1.92	$1''$	$2\frac{9}{16}''$	$1\frac{1}{2}''$	$3''$	$6''$	$12''$
$\frac{5}{8}''$	1.72	$1\frac{1}{16}''$	$2\frac{5}{8}''$	$1\frac{9}{16}''$	$3\frac{1}{8}''$	$6\frac{1}{4}''$	$12\frac{1}{2}''$

Triple-riveted Butt-joint with Outer and inner Welt-strips (Fig. 103).—This joint has three rows of rivets on each side of the butt. One row passes through the boiler-plate and one welt-strip and two rows pass through the sheet and two welts.

The resistance to tearing along line xx is

$$(2p - d)tf_t. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

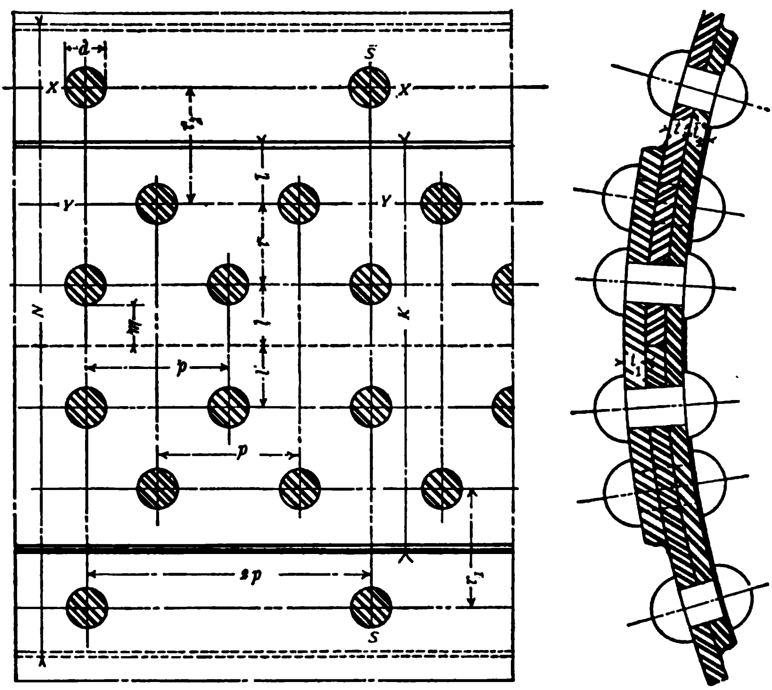


FIG. 103.

The resistance to pulling the plate out from between the welt-strips is

$$9 \times a \times f_s \times .85.$$

The resistance to tearing on line yy and shearing rivets on xx is

$$(2p - 2d)tf_t + 1af_s.$$

A glance at the figure will show that this joint cannot fail along the line zz, because there are two rivets in double shear and one rivet in single shear in addition to the net section of plate, which is equal to the net section on yy.

Exercise 36.—Make the drawings for a triple-riveted butt-joint like Fig. 103. Steel plates and iron rivets. $t = \frac{5}{8}''$, $d = 1\frac{1}{4}''$. *Scale 4″ = 1 foot.* The other dimensions may be taken from the following table:

TABLE 19.

t	d	p	m	r	K	N	Efficiency.
In.	In.	In.	In.	In.	In.	In.	Per Cent.
$\frac{3}{8}$	$\frac{13}{16}$	$2\frac{15}{16}$	$\frac{13}{16}$	$1\frac{11}{16}$	$8\frac{1}{4}$	$13\frac{5}{8}$	86.1
"	$\frac{7}{8}$	$3\frac{3}{16}$	$\frac{7}{8}$	$1\frac{13}{16}$	$8\frac{7}{8}$	$14\frac{1}{8}$	86.2
"	$\frac{15}{16}$	$3\frac{3}{8}$	$\frac{15}{16}$	$1\frac{15}{16}$	$9\frac{1}{2}$	$15\frac{1}{8}$	86.1
"	1	$3\frac{5}{8}$	1	$2\frac{1}{16}$	$10\frac{1}{8}$	$16\frac{1}{8}$	86.2
$\frac{7}{16}$	$\frac{7}{8}$	$3\frac{3}{16}$	$1\frac{7}{8}$	$1\frac{13}{16}$	$8\frac{7}{8}$	$14\frac{1}{8}$	86.2
"	$\frac{15}{16}$	$3\frac{3}{8}$	$\frac{15}{16}$	$1\frac{15}{16}$	$9\frac{1}{2}$	$15\frac{1}{8}$	86.1
"	1	$3\frac{5}{8}$	1	$2\frac{1}{16}$	$10\frac{1}{8}$	$16\frac{1}{8}$	86.2
"	$1\frac{1}{16}$	$3\frac{7}{8}$	$1\frac{1}{16}$	$2\frac{3}{16}$	$10\frac{3}{4}$	$17\frac{1}{4}$	86.2
$\frac{1}{2}$	1	$3\frac{5}{8}$	1	$2\frac{1}{16}$	$10\frac{1}{8}$	$16\frac{1}{8}$	86.2
"	$1\frac{1}{16}$	$3\frac{7}{8}$	$1\frac{1}{16}$	$2\frac{3}{16}$	$10\frac{3}{4}$	$17\frac{1}{4}$	86.2
"	$1\frac{1}{8}$	$4\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{5}{16}$	$11\frac{3}{8}$	$18\frac{1}{8}$	86.1
"	$1\frac{3}{16}$	$4\frac{5}{16}$	$1\frac{3}{16}$	$2\frac{7}{16}$	12	$19\frac{1}{8}$	86.2
$\frac{9}{16}$	$1\frac{1}{16}$	$3\frac{7}{8}$	$1\frac{1}{16}$	$2\frac{3}{16}$	$10\frac{3}{4}$	$17\frac{1}{8}$	86.2
"	$1\frac{1}{8}$	$4\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{5}{16}$	$11\frac{3}{8}$	$18\frac{1}{8}$	86.1
"	$1\frac{3}{16}$	$4\frac{5}{16}$	$1\frac{3}{16}$	$2\frac{7}{16}$	12	$19\frac{1}{8}$	86.2
"	$1\frac{1}{4}$	$4\frac{9}{16}$	$1\frac{1}{4}$	$2\frac{5}{8}$	$12\frac{3}{4}$	$20\frac{1}{4}$	86.3
$\frac{5}{8}$	$1\frac{3}{16}$	$4\frac{5}{16}$	$1\frac{3}{16}$	$2\frac{3}{16}$	12	$19\frac{1}{8}$	86.2
"	$1\frac{1}{4}$	$4\frac{9}{16}$	$1\frac{1}{4}$	$2\frac{5}{8}$	$12\frac{3}{4}$	$20\frac{1}{4}$	86.3
"	$1\frac{5}{16}$	$4\frac{3}{4}$	$1\frac{5}{16}$	$2\frac{3}{4}$	$13\frac{3}{8}$	$21\frac{1}{4}$	86.1
$\frac{11}{16}$	$1\frac{5}{16}$	$4\frac{3}{4}$	$1\frac{5}{16}$	$2\frac{3}{4}$	$13\frac{3}{8}$	$21\frac{1}{4}$	86.1
"	$1\frac{3}{8}$	5	$1\frac{3}{8}$	$2\frac{7}{8}$	14	$22\frac{1}{4}$	86.2
$\frac{3}{4}$	$1\frac{3}{8}$	5	$1\frac{3}{8}$	$2\frac{7}{8}$	14	$22\frac{1}{4}$	86.2

Calculate efficiency and if possible show where improvement might be made.

Exercise 37.—Draw the junction of a longitudinal double-riveted butt-joint with a single-riveted circumferential lap-joint (Fig. 104). $t = \frac{5}{8}''$, $d = \frac{13}{16}''$. The remaining dimensions may be taken from Tables 18 and 14. *Scale 6″ = 1 foot.*

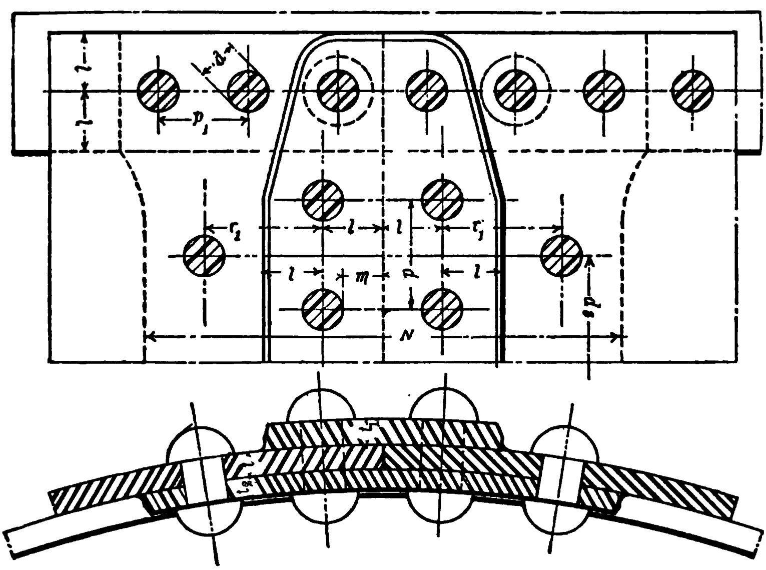


FIG. 104.

Exercise 38.—Make drawings of the staying for the back-head and fire-box crown-sheet of a locomotive-boiler as shown by Fig. 105. *Scale 3″ = 1 foot.*

This is an example of what is known as the *crown-bar* staying for locomotive-boilers. The design is suitable for an engine with cylinders 19″ × 24″, steam-pressure 180 lbs. per square inch, and is similar to that used in the Empire State Express locomotive designed by Wm. Buchannan, Supt. of Motive Power of the N. Y. C. R. R. *AA* shows a cross-section and a partial elevation of one crown-bar which consists of two

wrought-iron plates 5″ deep × $\frac{3}{4}$″ thick and welded together at the ends. The fire-box crown-sheet is supported by $\frac{7}{8}$″ rivets, which, passing through a washer *b* and between the

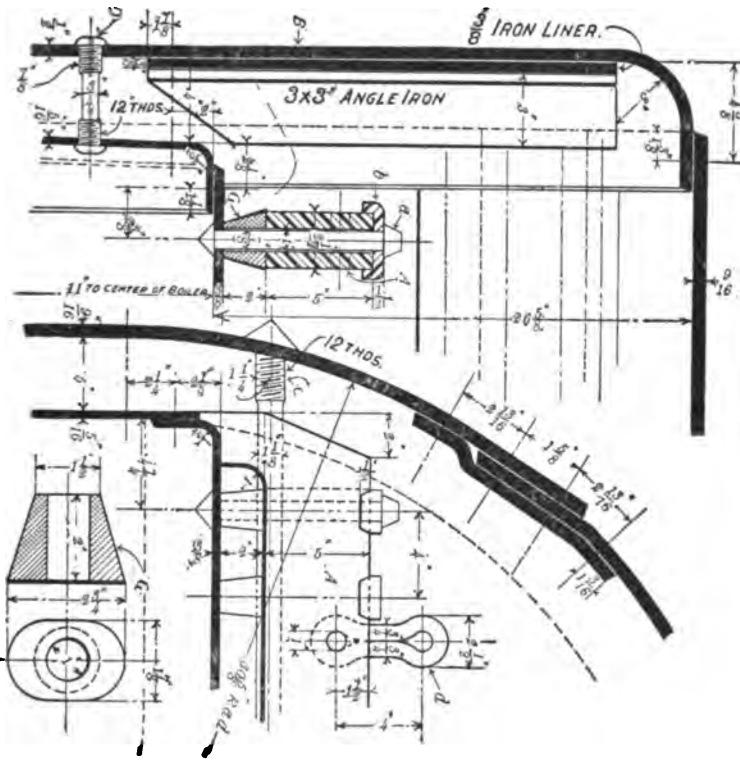


FIG. 105.

plates *A* of the bar and through thimble *G*, is riveted on the under side of the crown-sheet as shown. These rivets are placed from 4″ to 4½″ apart, and as many as the crown-bars will accommodate at these centres, the end bolts being placed about 4″ from the inside of the fire-box side sheets. As seen from the figure, the crown-bars are placed in a transverse position

on the crown-sheet, and as many as the longitudinal length of the sheet will allow, with equal spacing, about $4\frac{1}{2}''$ apart. Should these bars be insufficient to support the crown-sheet against the downward pressure of the steam, which is equal to the area of the crown-sheet $\times$ the steam-pressure per square inch, then what remains is held up by *sling*-stays hung from the outer shell and fastened to the crown-bars by links and pins, one link of which is shown at *d* in the transverse cross-section.

The flat upper part of the back-head, which has no stay-bolts passing through it like those which bind the fire-box and outer shell together, as shown at *D*, is stiffened with a liner $\frac{3}{8}''$ thick, the shape of which is shown by dotted lines on the transverse section, and to this liner are riveted as many lengths of $3'' \times 3''$ angle-iron as can be placed on the liner, with a clearance-space of only about $\frac{3}{4}''$ between. To these angle-irons are bolted longitudinal stay-rods $1\frac{1}{8}''$ in diameter similar to that shown in Fig. 106.

To support that curved part of the outside shell just above the fire-box transverse stay-rods *C* are carried between each crown-bar, screwed through the shell on each side, and riveted over on the outside. The body of the rod is $1\frac{1}{8}''$ in diameter and the screwed ends $1\frac{1}{4}''$ diameter.

The fire-box stay-bolts *D* are screwed through both fire-box and outer shell and riveted over outside and inside. It will be seen that while the screwed part of the bolt is $\frac{7}{8}''$ diameter the body is turned down to $\frac{3}{4}''$, which reduces its stiffness and allows it to give somewhat to the unequal expansion of the fire-box and outer shell of the boiler. In certain places the stay-bolts are more liable to break than in others; in such

places hollow stay-bolts are used, so that when broken they may be easily and quickly detected.

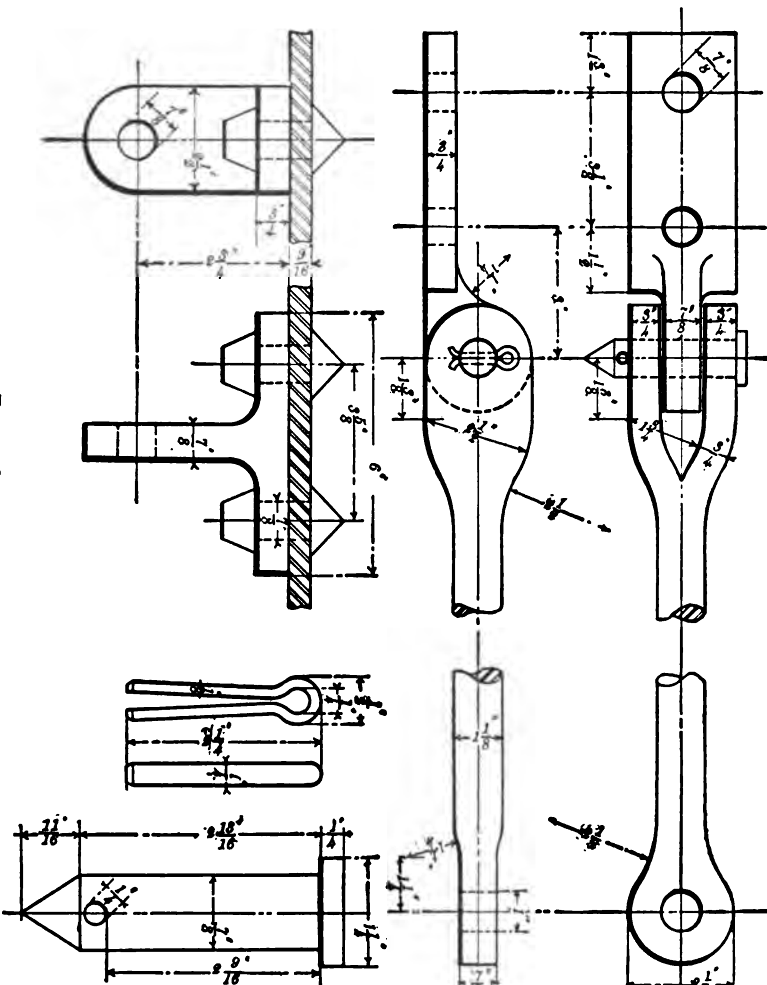


FIG. 106.

Hollow stay-bolts have an $\frac{1}{8}''$ hole drilled completely through from fire-box to the outside of the outer shell, so that

if one should break the escaping steam and water will soon inform the engineer.

A detail view of one of the crown-bar thimbles is shown at *G*.

Construction.—Draw the perpendicular centre line 6′ 6″ from the left-hand margin, and the longitudinal centre line 4′ 6″ from the upper margin; then draw the transverse and

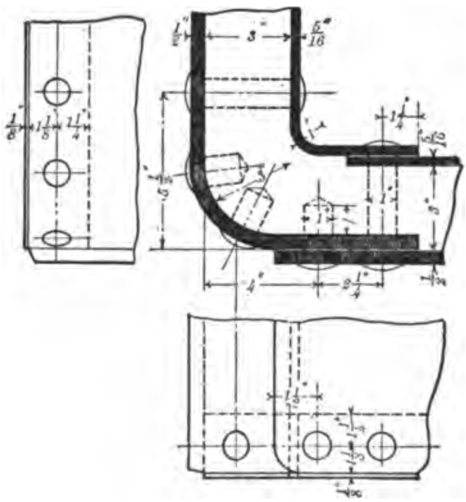


FIG. 107.

longitudinal cross-sections of the boiler and construct the crown-bars and other staying as shown.

It will be seen in the figure that where the plates should come together they have been left slightly apart; this is a convention followed by draftsmen to facilitate inking without blots and to improve the appearance of the drawing.

Exercise 39.—Fig. 106 gives an example of a *longitudinal stay-rod* with details and a *crow-foot* for a locomotive-boiler. Make the drawings to a *scale of 6″ = 1 foot.*

Exercise 40.—Figs. 107 and 108 show examples of riveting the corner of a locomotive fire-box ring (sometimes called a

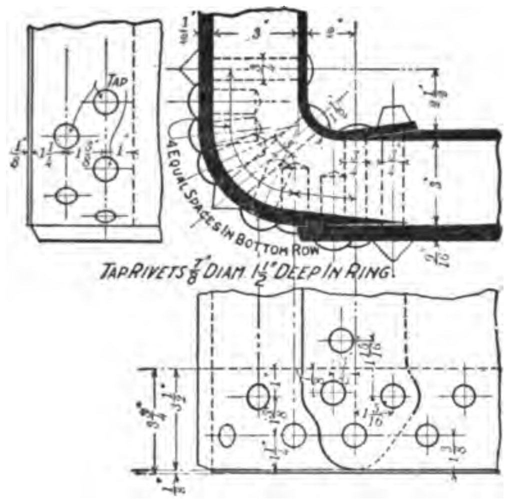


Fig. 108.

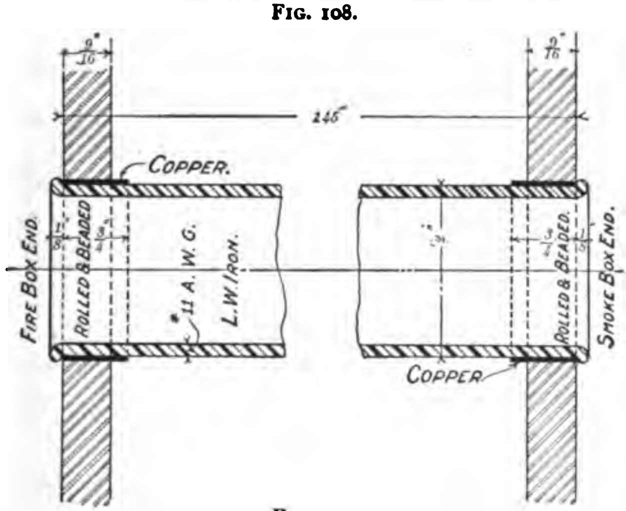


Fig. 109.

mud-ring) to the bottom of the fire-box and outer shell of the boiler. Fig. 108 is that of a large boiler 58″ diameter

and carrying 180 lbs. steam-pressure per square inch, and Fig. 107 is for a smaller boiler of 48″ diameter at waist. Make drawings of both figures as shown to a *scale of 4″ = 1 foot.*

Exercise 41.—Fig. 109 shows the setting of a tube in the front and back tube-sheets of a fire-tube boiler for a locomotive. Both ends show the tubes swedged, rolled, and beaded, and with copper ferrules between the tubes and the sheets. Make a drawing like that shown by the figure to the *scale of full size.*

Fig. 110 is a section of a locomotive-boiler dome, dome-ring C and dome-base B. The base and ring are made of soft steel and formed in dies by hydraulic pressure.

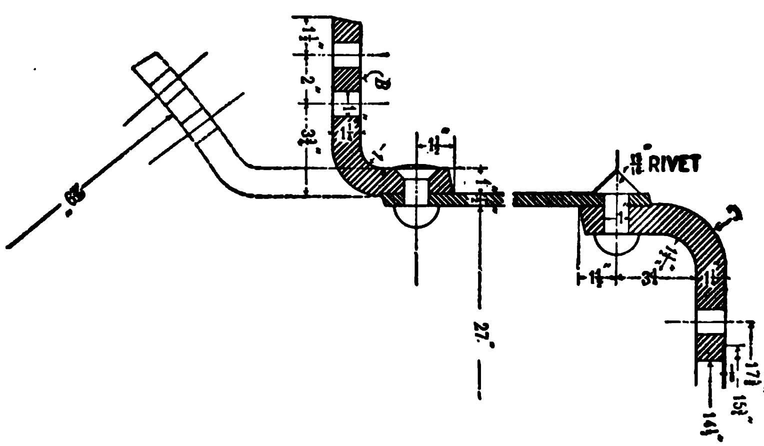


FIG. 110.

Exercise 42.—Make *half sectional elevation, half outside elevation* with *transverse view* and plan. Also show the curves of intersection between the dome-base and boiler. *Scale 2″ = 1 foot.*

CHAPTER IV.

SHAFTING AND SHAFT-COUPLINGS.

UNDER the term *shafting* may be included *line shafting* and *axles*.

Line Shafting.—This name is given to the long line of rotating, cylindrical or square shafting used in workshops and factories for transmitting turning power or *twisting moments* from the prime movers. They are in some ways an extension of the prime mover. Such shafting is subjected to torsional and bending stresses, the latter being due to the pull of belts and the weight of pulleys, gears, levers, etc. "It is usual to make line shafting of uniform diameter throughout, as shown in Fig. 111, enlarged ends being only used occasionally for

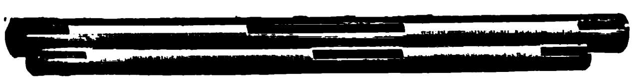


FIG. 111.

exceptional purposes. *Steel* of a grade containing .3 to .4% of carbon is now used almost entirely for shafting in preference to *iron* in this country. The commercial lengths of shafting for ordinary diameters, as from 2″ to 3″, run from 16 ft. to 30 ft., the shorter lengths being more convenient for transportation, for replacing pulleys, gears, etc. But the

longer lengths are frequently used when objections do not arise from these considerations." *

Torsional or Twisting Moment.—Figs. 112 and 113 show a lever, a gear wheel and pinion keyed to their respective shafts; R is the radius of the lever and the pitch circle of the gear through which the power P is transmitted. This force P produces a twisting action on the shaft, and the product RP is called the *torsional moment* (T) on the shaft.

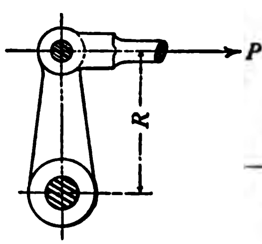


FIG. 112.

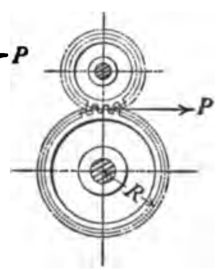


FIG. 113.

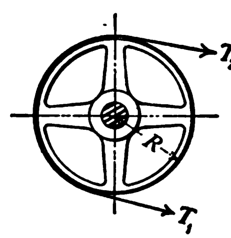


FIG. 114.

So in Fig. 114 P is equal to the tension $T_1 - T_2$, and the radius R multiplied by the force P is again equal to the torsional moment on the shaft. The torsional moment is usually expressed in inch-pounds, i.e., the force P in pounds into the radius R in inches is equal to the torsional moment in inch-pounds.

The moment of resistance to torsion of a cylindrical shaft is equal to the greatest stress multiplied by the modulus of the section.

Let F_t be the greatest shearing stress and Z_t the modulus: then

$$T = F_t Z_t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

* A. & P. Roberts Company.

and $$Z_t = \frac{\pi}{16}d^3 = .19635d^3;$$

so for cylindrical shafts

$$T = .19635d^3f_s, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and for square shafts

$$T = .208d^3f_s; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$d =$ diameter of the cylindrical shaft and length of side of square shaft in inches;

$f_s =$ shearing strength in pounds per square inch;

$T =$ torsional moment in inch-pounds.

To Find the Diameter of a Wrought-iron or Steel Shaft.—If we take the resistance to shearing for iron equal to 40,000 lbs. per square inch and for soft steel at 50,000 lbs., and using a factor of safety of 4½, we have:

For cylindrical iron shafts $T = 1720d^3$;
For cylindrical steel shafts $T = 2182d^3$;
For square iron shafts $T = 1849d^3$;
For square steel shafts $T = 2311d^3$.

Then

$$d = \begin{cases} \sqrt[3]{\dfrac{T}{1720}} \text{ for cylindrical iron shafts;} \quad . \quad . \quad (4) \\ \sqrt[3]{\dfrac{T}{1849}} \text{ for square iron shafts;} \quad . \quad . \quad . \quad (5) \\ \sqrt[3]{\dfrac{T}{2182}} \text{ for cylindrical steel shafts;} \quad . \quad (6) \\ \sqrt[3]{\dfrac{T}{2311}} \text{ for square steel shafts.} \quad . \quad . \quad . \quad (7) \end{cases}$$

Example 1.—Let the pitch diameter of a spur gear on an iron shaft be 60″, and the total pressure on the teeth at the pitch line 2500 lbs. What would be the diameter of the shaft and the horse-power transmitted by the wheel if running at the rate of 140 revolutions per minute?

From equation (4) we get

$$d = \sqrt[3]{\frac{2500 \times 30}{1720}} = 3\tfrac{1}{2}''$$

for the horse-power transmitted by the wheel.

Let H.P. = horse-power = 33,000 × 12 = 396,000 inch-pounds;

n = number of revolutions per minute;

then

$$\text{H.P.} = \frac{2\pi R \times P \times n}{396,000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$= \frac{6.28 \times T \times n}{396,000} = \frac{6.28 \times 75,000 \times 140}{396,000} = 166.5.$$

In terms of the H.P.,

$$T = \frac{63,057 \text{ H.P.}}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and

$$d = \sqrt[3]{\frac{36 \text{ H.P.}}{n}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Besides the twisting stresses on shafts which we have alone taken account of in the above formulæ, there is usually a bending moment to be considered. Let a shaft be subjected

to a torsional moment T and supporting a bending moment B; these two stresses will be equal to a twisting moment

$$T_1 = B + \sqrt{(B^2 + T^2)}. \quad . \quad . \quad . \quad . \quad (11)$$

T_1 is called the *equivalent* twisting moment, and should be used in place of T in figuring the diameter of a shaft subjected to combined torsion and bending. The bending stresses in revolving shafts are continually changing from tension to compression and from compression to tension, so that for combined bending and twisting the factor of safety $4\frac{1}{2}$ given for twisting alone should be increased in the following ratio: When B is more than $.3T$ and not more than $.6T$, the factor of safety should be 5; when over $.6T$ and not more than T, $5\frac{1}{2}$; and when greater than T, 6.

Example 2.—Determine the diameter of a wrought-iron shaft which has to resist a torsional moment of 400,000 inch-pounds and a bending moment of 200,000 inch-pounds. By formula (11) the equivalent twisting moment

$$T_1 = B + \sqrt{B^2 + T^2} = 200{,}000 + \sqrt{200{,}000^2 + 400{,}000^2}$$
$$= 200{,}000 + 538{,}145 = 738{,}145 \text{ in.-lbs.};$$

and by equation (4)

$$d = \sqrt[3]{\frac{T_1}{1720}} = \sqrt[3]{\frac{738{,}145}{1720}} = 7\frac{1}{2}''.$$

Example 3.—When the bending moment exceeds the torsional moment.—A non-continuous steel shaft has its bearings 8 ft. apart and carries a pulley of 50″ diameter at its centre; the pulley is driven by a 10″ belt, the effective weight and

belt-pull being 500 and 800 lbs. respectively. What should be the diameter of the shaft?

In this case the factor of safety will be 6 and equation (4) becomes

$$d = \sqrt[3]{\frac{T_1}{1634}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$B = \frac{800 + 500 \times 96}{4} = 31,200 \text{ inch-lbs.};$$

$$T = 800 \times 25 = 20,000 \text{ inch-lbs.};$$

$$T_1 = 31,200 + \sqrt{31,200^2 + 20,000^2} = 68,230;$$

and

$$d = \sqrt[3]{\frac{68,230}{1634}} = 3\tfrac{1}{2}'' \text{ nearly.}$$

Deflection of Shafting.—A maximum deflection of $\frac{1}{100}$ of an inch per foot of length l for continuous shafting is given as good practice by the Pencoyd Iron Works. The weight of bare shafting $= 2.6d^2 \times l = W$, and for loaded shafts, allowing 40 lbs. per inch of width for the vertical pull of the belts, $W = 13d^2l$. Then for bending stress alone, taking the modulus of transverse elasticity at 26,000,000, we can derive from authoritative formulæ the maximum length between bearings

$$l = \sqrt[3]{873d^2} \text{ for bare shafts}; \quad . \quad . \quad . \quad . \quad (13)$$

$$l = \sqrt[3]{175d^2} \text{ for loaded shafts.} \quad . \quad . \quad . \quad (14)$$

For line-shafting hangers 8 ft. apart Thurston gives

$$\text{H.P.} = \frac{d^3 n}{90} \quad d = \sqrt[3]{\frac{90\ \text{H.P.}}{n}} \quad \text{for wrought iron;}$$

$$\text{H.P.} = \frac{d^3 n}{55} \quad d = \sqrt[3]{\frac{55\ \text{H.P.}}{n}} \quad \text{for cold-rolled iron.}$$

Hollow Shafts.—Weight for weight the hollow shafts are stronger than solid shafts, because the portion of material removed is the least effective in resisting torsion. The resistance to torsion in a solid shaft and a hollow shaft will be equal when the moduli of the sections are equal. Let d be the diameter of a solid shaft, and d_1 and d_2 the internal and external diameters respectively of a hollow shaft. Then

$$d^3 = \frac{d_2^4 - d_1^4}{d_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

A 10″ hollow shaft with hollow 4″ diameter will weigh 16% less than the solid 10″ shaft, but its strength will be only 2.56% less. If the hole were increased to 5″ diameter the weight would be 25% less than that of the solid shaft, and the strength only 4.25% less.

The relation between the weights of solid and hollow shafts is as follows:

Let W = weight of hollow shaft, and W_1 = weight of solid shaft; then

$$\frac{W}{W_1} = \frac{d_2^2 - d_1^2}{d^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

the weight of the hollow shaft in per cent of the weight of the solid shaft.

And the difference in strength is given as follows:

Let S = strength of hollow shaft, and S_1 = strength of solid shaft; then

$$\frac{S}{S_1} = \frac{d_2^4 - d_1^4}{d_2 d^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (17)$$

When $d^2 = d_2^2 - d_1^2$ the solid and hollow shafts are of equal weight; and when $d^3 = \dfrac{d_2^4 - d_1^4}{d_2}$ the solid and hollow shafts are of equal strength.

A hollow shaft is stiffer than a solid shaft of the same resistance to torsion, i.e., it does not yield to bending as readily; this is an objection under some conditions, as in the case of a steamship's propeller-shaft, where it would be an advantage if the shaft would *give* a little to the straining action of the ship in a storm.

The larger hollow shafts are usually forged hollow, but the smaller shafts are forged or rolled solid and then bored hollow.

SHAFT-COUPLINGS.

We have seen that shafting is made only in short lengths of about 16 ft. to 30 ft. long, so it is necessary in long lines of shafting to couple these lengths together by means of what is known as shaft-couplings. There are three principal divisions of shaft-couplings, viz., *rigid*, *flexible*, and *clutch couplings*. All couplings should be placed close to bearings on the side farthest from the driving-point.

Rigid Couplings.—When two shafts are joined by a coupling that can only be removed by loosening keys or

unscrewing bolts, such a coupling is said to be a rigid or *fast* coupling.

Box or Muff Couplings.—This is the simplest kind of a shaft-coupling (Fig. 115). It is made of *cast iron*. The hole for the shaft is cored small in the casting and afterwards bored out to fit the finished diameter of the shaft. The coupling is secured to the shaft by means of a wrought-iron or steel sunk key about equal in length to the coupling itself, or by two keys each about half as long as the coupling. The latter method is the best, because then it is not so necessary that the keyway in both shafts should be exactly the same depth; moreover, the two keys can be driven tighter and slacked easier than one long key. Half the depth of the keyway is cut in the shaft and half in the coupling. The two shafts butt together at the ends. When two keys are used a clearance space should be left between them when driven home, to insure an equal tightness of both keys, as shown at *S*, Fig. 115.

Exercise 43.—Make a complete working drawing of a muff coupling like Fig. 115 for a $2\frac{1}{2}''$ shaft. *Scale = full size.*

The dimensions may be found from the following proportions:

$d =$ diameter of shaft $= 2\frac{1}{2}''$;
$t =$ thickness of metal in coupling $= .4d + \frac{1}{2}''$;
$l =$ length of muff $= 2\frac{1}{2}d + 2''$;
$D = d + 2t$.

For proportions and taper of key see page In some positions of the coupling on the shaft the key should have a *gib head*, as shown at *h* in Fig. 117, when it is difficult to

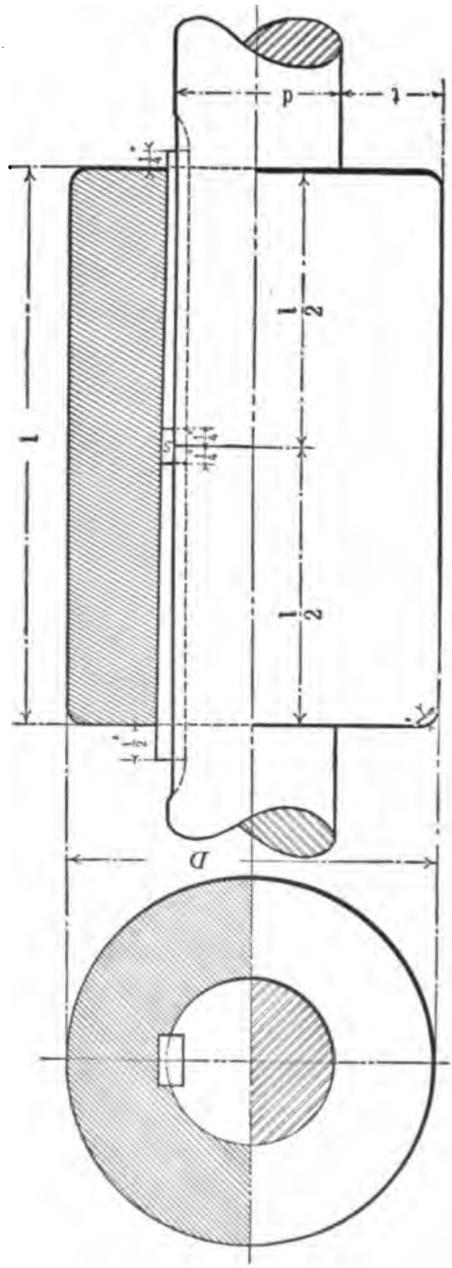


FIG. 115.

obtain access to the small end for the purpose of slacking up. When the coupling is situated close to the bearing it will be necessary to make the length of the keyway to the left of the coupling equal to half that of the corresponding key.

Split-muff Couplings.—Fig. 116 shows a form of coupling divided into two parts and bolted together on the shaft, sometimes called a *compression* coupling. It is keyed to the shaft with a straight parallel key which fits only at its sides. The length of the key may be $\frac{1}{8}''$ longer than the coupling and the keyway the same length as the key.

Exercise 44.—Make working drawings of the split-muff coupling shown in Fig. 116 for a 3″ shaft. *Scale 6″ = 1 foot.*

In finishing this coupling the inside faces of the two halves are planed and the bolt-holes drilled. Now placing a piece of sheet tin between the halves and bolting them together, the hole is bored for the shaft, making it equal in diameter to the finished shaft. The sheet tin is then removed, and the coupling when bolted on the ends of two shafts clamps them very tightly together. The ease with which the split coupling can be removed and replaced gives it a great advantage over the solid-muff coupling.

Flange or Plate Coupling.—Fig. 117 shows a plate coupling made by the Dodge Mfg. Co. It is made in two parts, of cast iron, and keyed to the two shaft ends, the position of the key in the one shaft being at right angles to the key in the other. The bolts are turned and carefully fitted, and the holes drilled to template so that they can be duplicated if desired. Each coupling should be faced after it has been keyed to its shaft, so as to obtain perfect alignment.

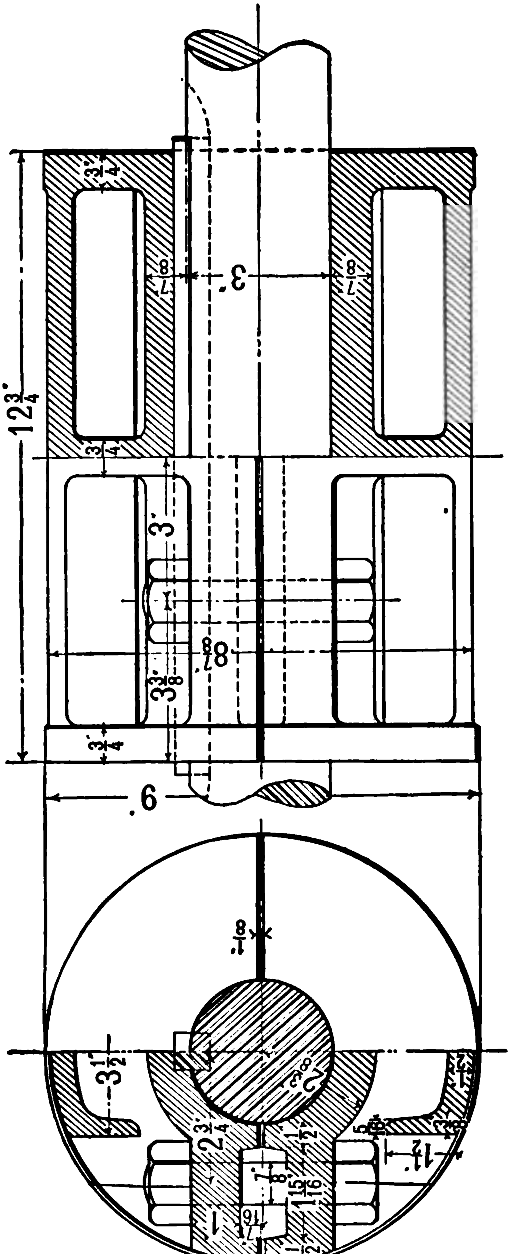


FIG. 116.

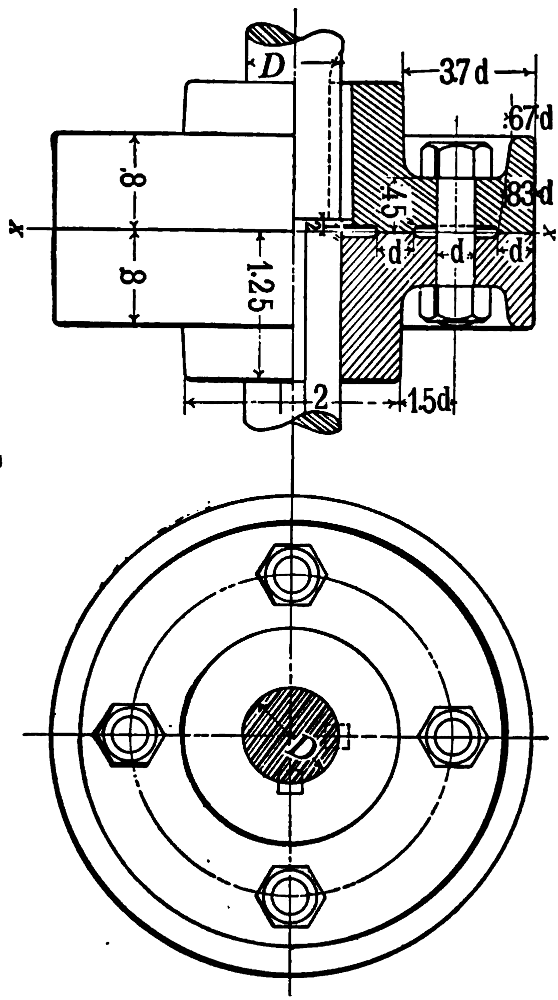


Fig. 117.

Exercise 45.—Make drawings of Fig. 117 as shown, except that the upper half of the end elevation shall be a *sectional* view through XX.

Let $D =$ diameter of shaft $= 2''$;

Unit $= D + \frac{1}{2}''$;

$$n = \text{number of bolts} = 3 + \frac{D}{2}. \quad . \quad . \quad . \quad . \quad . \quad (18)$$

Taking the nearest even number,

$$d = \text{diameter of bolt} = \frac{D}{n} + \frac{1}{4}. \quad . \quad . \quad . \quad (19)$$

FIG. 118.

The remaining dimensions can be found from the proportions given in the figure in terms of d, the diameter of the bolt.

The taper of the hub may be made equal to $\frac{1}{8}''$ in $12''$.

The shaft in this figure is shown sectioned for wrought iron, but in the drawing required it may be sectioned with steel *color.*

Figs. 118 and 119 show plate couplings made by the Hill Clutch Company.

Exercise 46.—Make drawings as required for Exercise 45 of a "Hill" plate coupling for a $4\frac{1}{2}''$ shaft. The dimensions for *A* and *B* to be taken from Table 20. The number of

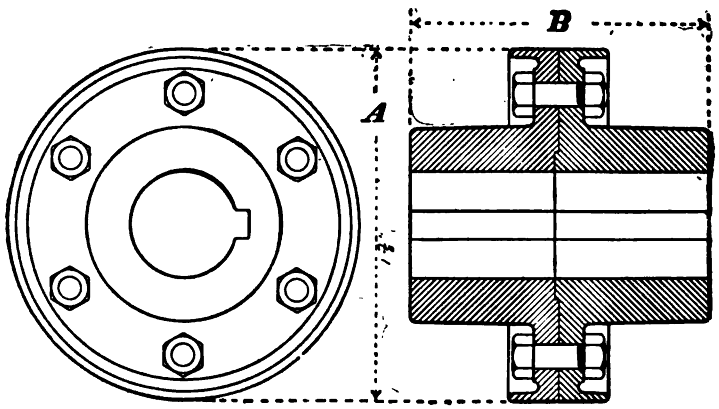


FIG. 119.

TABLE 20.

DIMENSIONS FOR THE "HILL" PLATE COUPLINGS.

Diameter Shaft.	A	B	Diameter Shaft.	A	B	Diameter Shaft.	A	B
$1\frac{3}{16}$	7	6	$2\frac{15}{16}$	$11\frac{1}{2}$	$9\frac{1}{2}$	$5\frac{1}{2}$	$17\frac{1}{2}$	16
$1\frac{7}{16}$	7	6	$3\frac{3}{16}$	$11\frac{1}{2}$	10	6	20	17
$1\frac{11}{16}$	8	6	$3\frac{7}{16}$	$12\frac{1}{2}$	$10\frac{1}{2}$	$6\frac{1}{2}$	21	$18\frac{1}{2}$
$1\frac{15}{16}$	$8\frac{1}{2}$	$6\frac{1}{2}$	$3\frac{11}{16}$	$12\frac{1}{2}$	$10\frac{1}{2}$	7	22	20
$2\frac{3}{16}$	$8\frac{1}{2}$	$6\frac{1}{2}$	$3\frac{15}{16}$	13	$11\frac{1}{2}$	8	24	22
$2\frac{7}{16}$	10	8	$4\frac{7}{16}$	$14\frac{1}{2}$	$12\frac{1}{2}$	9	26	24
$2\frac{11}{16}$	11	$8\frac{1}{2}$	5	$16\frac{1}{2}$	$14\frac{1}{2}$	10	28	26

bolts to be determined from equation (18), and the diameter of the bolts from equation (19) The remaining proportions to be worked out according to the student's judgment.

The Sellers Clamp Coupling (Fig. 120).—This is a special form of a muff coupling which is turned to a cylin-

drical form on the outside, but has a double conical section inside. Two conical sleeves or bushes turned to fit the

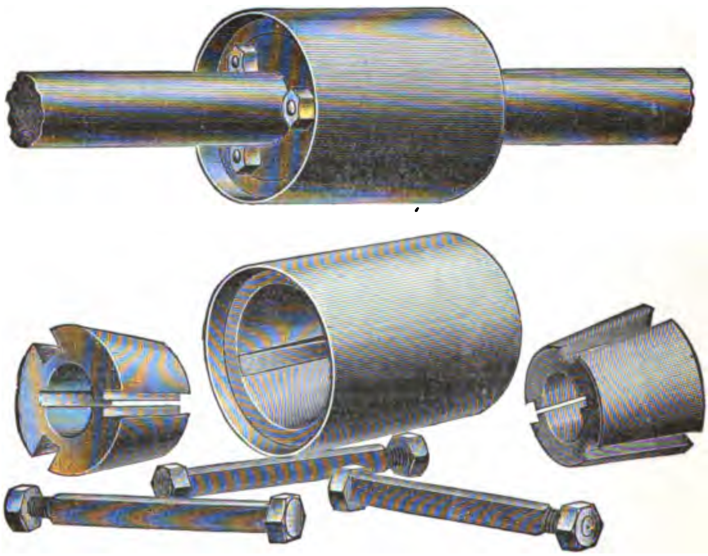


FIG. 120.

inside of the muff and bored out to fit the shafts are pulled together by three bolts. The sleeves are split on one side through one of the bolt-holes, so that the more the bolts are screwed up, the tighter the sleeves clamp the shafts and bind them firmly together. Keys are also used to further prevent slipping.

Exercise 47.—Make the drawings shown in Fig. 121 of a Sellers clamp coupling. *Scale full size.*

The taper of the conical sleeve is 2¾″ per foot of length *on the diameter;* e.g., if the sleeve was 6″ long and the large diameter measured 4″, the small diameter would measure 2⅝″.

For the dimensions of the Sellers clamp coupling for various diameters of shaft, use the following table.

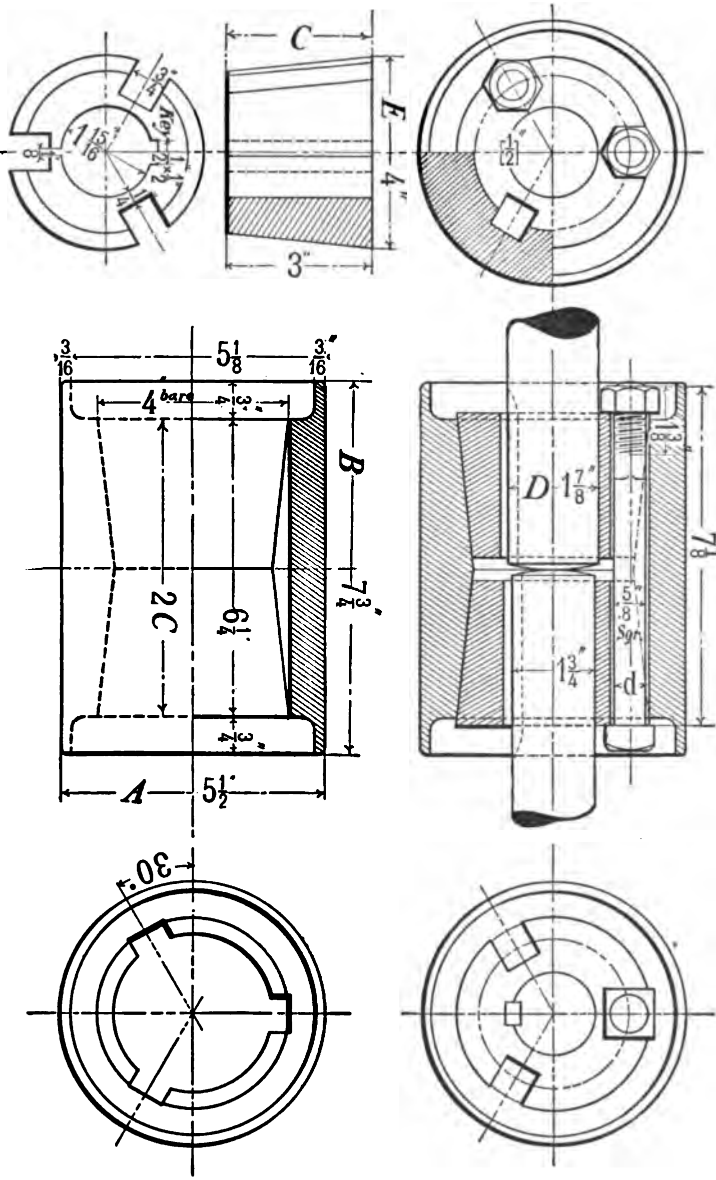


FIG 121.

TABLE 21.

SELLERS CLAMP COUPLINGS.

D	*A*	*B*	*C*	*E*	*d*	*D*	*A*	*B*	*C*	*E*	*d*
1½″	4½″	5⅝″	2¼″	3½	$\frac{7}{16}$″	3″	8½″	11⅜″	4½″	6¾″	⅞″
1¾	5¾	6⅜	2⅝	4	½	3½	9¾	13⅛	5¼	7¾	1
2	6⅛	7¾	3	4¾	⅝	4	11	14⅞	6	8¾	1⅛
2¼	6½	8½	3⅜	5	⅝	5	12⅞	18¼	7½	10¼	1¼
2½	7¼	9½	3¾	5¾	¾	6	14½	21¼	9	11¼	1¼
2¾	7⅝	10¼	4⅛	6	¾						

Frictional Coupling.—Fig. 122 shows three views of Butler's frictional coupling. It is somewhat like the Sellers coupling, except that it has neither bolts nor keys, the conical bushes being held in position by round nuts threaded into the muff. The conical bushes are split at the side, and when they are in position on the shaft the split sides are at right angles to each other; this arrangement allows a key-driver to be introduced through one of these openings (after the nuts have been removed) to drive out the other bush when it is desired to remove the coupling from the shaft. The bushes are guided into position by small dowel-pins which enter short grooves provided for them inside the muff. The ⅜″ round holes shown in top and bottom at the centre of the muff are used to see when the ends of the shafts come together, for then only will the coupling be in its proper position.

Exercise 48.—Make complete working drawings of the Butler coupling like Fig. 122, except that the shaft shall be of steel and the sectioning shall be appropriately colored instead of hatch-lined. *Scale = full size.*

The threads on the lock-nuts should be that number per inch used on a pipe whose *outside* diameter is nearest to the

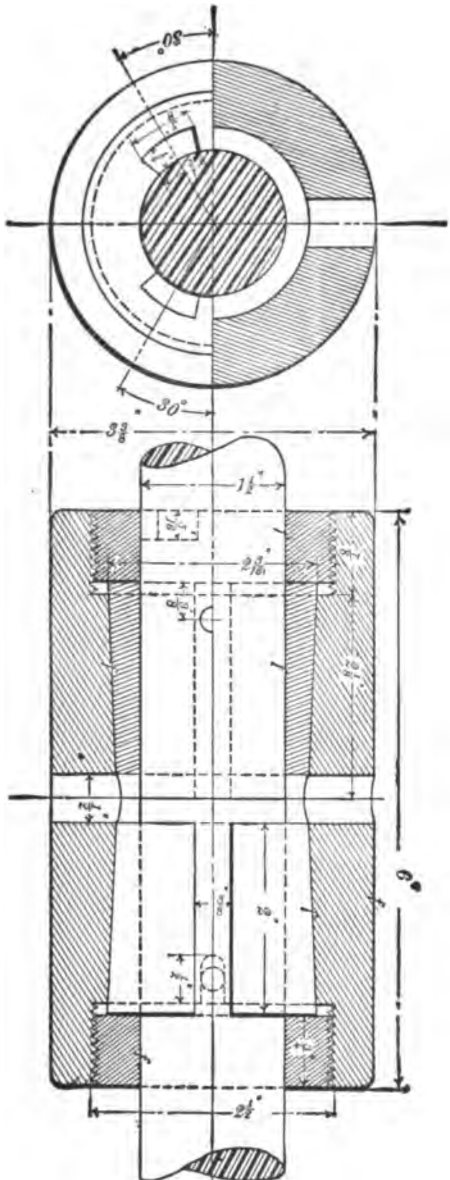


FIG. 122.

outside diameter of the nut. The lock-nuts are screwed into position by means of a spanner wrench having projecting pieces which fit into the recesses shown in end elevation. The taper of the conical bushes may be made $\frac{3}{4}''$ in $12''$ *on the diameter.* The faces marked with small f are to be finished.

The principal proportions of this coupling are as follows:

$d =$ diameter of shaft;

$D =$ diameter of muff $= 2.25d$;

$L =$ length of muff $= 4d$.

Stuart's Clamp Coupling.—This coupling, shown in Fig. 123, differs from the Sellers coupling in having tapered

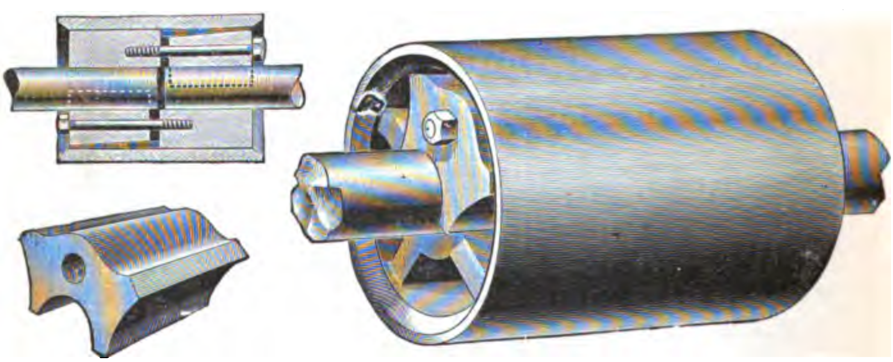


FIG. 123.

wedges instead of conical sleeves; these tapered wedges and opposite halves of each end of the muff are bored to the size of the shaft. Studs and nuts hold the wedges in place, making, on the whole, a cheap and effective coupling without the use of keys.

Exercise 49.—Make drawings of a Stuart's coupling as shown in Fig. 124 for a $1\frac{5}{8}''$ shaft. *Scale = full size.*

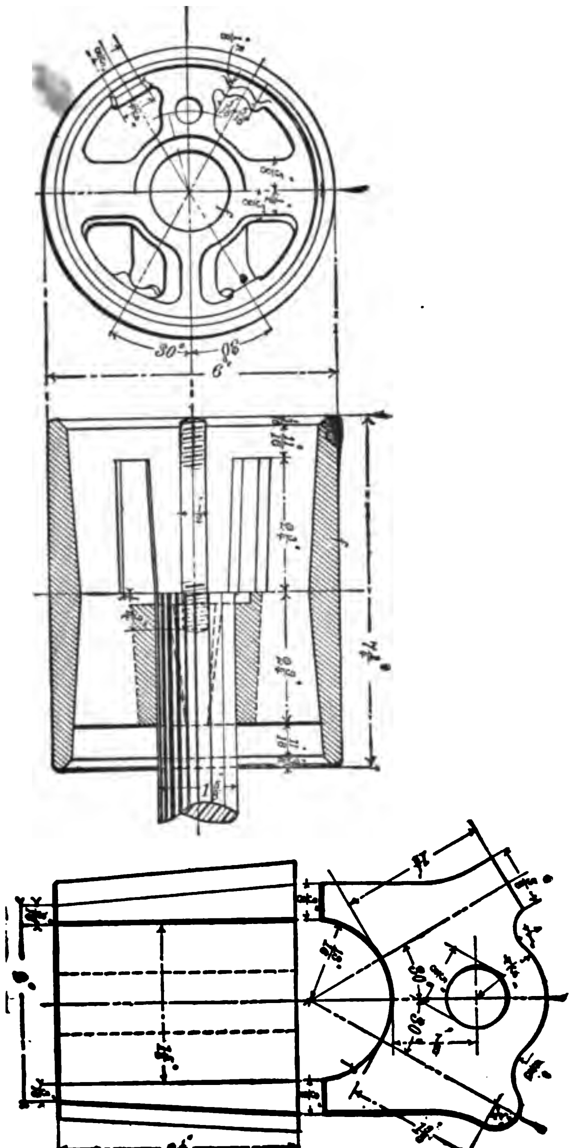


FIG. 124.

The principal dimensions of this coupling for various diameters of shaft are given in the following proportions:

Let $d =$ diameter of shaft;

$D =$ diameter of muff;

$L =$ length of muff.

Then for shafts from $1\frac{1}{4}''$ to $2\frac{3}{4}''$ diameter

$$D = 3.25d, \qquad L = 4.25d;$$

for shafts from $2\frac{3}{4}''$ up

$$D = 3d, \qquad L = 4d.$$

Flanged Shaft-coupling.—Fig. 125 shows a propeller-shaft coupling in which the flanges are formed by forging them on the shaft.

Exercise 50.—Make working drawings of the flange coupling shown in Fig. 125. Assume the external diameter of the shaft to be 18″ and the internal diameter 10″, and take an equivalent solid shaft as the unit for the proportions. *Scale 2″ = 1 foot.*

Let D_1 and D_2 be the internal and the external diameter, respectively, of the hollow shaft; then from equation (15) we have

$$D = \sqrt[3]{\frac{D_2^4 - D_1^4}{D_2}}$$

= diameter of an equivalent solid shaft = unit.

$d =$ diameter of bolt;

$n =$ number of bolts $= .25D + 2$;

$$R = \text{radius of bolt circle} = \frac{D + 2.25d}{2}.$$

Resistance to shearing of bolts = resistance to torsion of shaft divided by R.

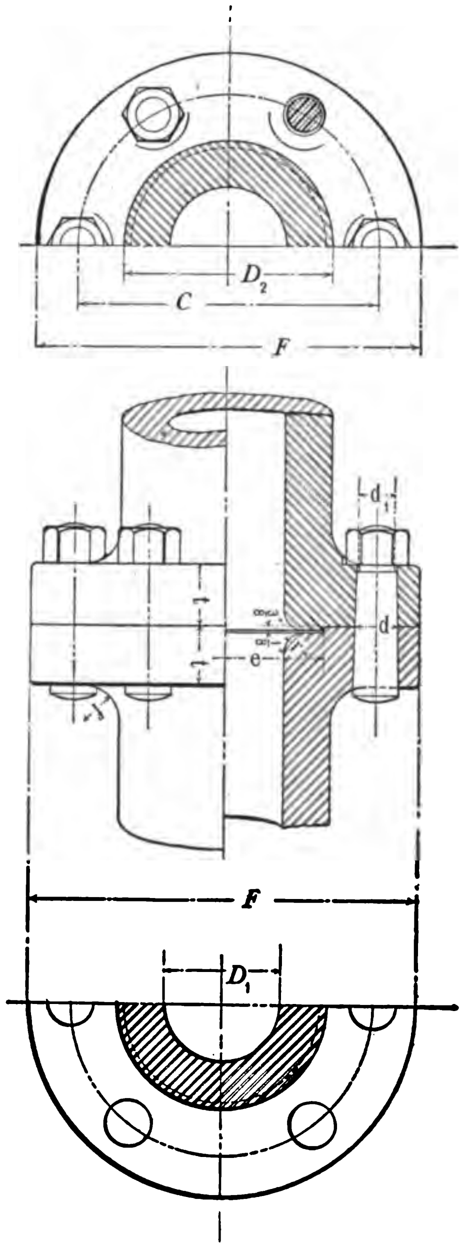


FIG. 125.

From equation (2),

$$.7854 d^2 n f R_s = .196 D^3 f_s; \quad . \quad . \quad . \quad . \quad (20)$$

and taking f_s at 50,000 for the steel shaft and 40,000 for the wrought-iron bolts, and using a factor of safety of 5, we have

$$d = 0.55\sqrt{\frac{D^3}{nR}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

It is evident that we must find R before we can determine d. The following table, by D. A. Low, gives values of d and n for solid shafts:

TABLE 22.

FLANGED SHAFT-COUPLINGS.

D	n	d	n	d	D	n	d	n	d
3	3	$\frac{15}{16}$	4	$\frac{7}{8}$	14	6	$3\frac{5}{16}$	8	$2\frac{15}{16}$
4	3	$1\frac{1}{4}$	4	$1\frac{1}{8}$	15	6	$3\frac{9}{16}$	8	$3\frac{1}{8}$
5	4	$1\frac{7}{16}$	6	$1\frac{3}{16}$	16	6	$3\frac{13}{16}$	8	$3\frac{3}{8}$
6	4	$1\frac{11}{16}$	6	$1\frac{7}{16}$	17	6	$4\frac{1}{16}$	8	$3\frac{9}{16}$
7	4	2	6	$1\frac{11}{16}$	18	6	$4\frac{5}{16}$	8	$3\frac{3}{4}$
8	4	$2\frac{1}{4}$	6	$1\frac{7}{8}$	19	8	4	9	$3\frac{13}{16}$
9	6	$2\frac{1}{8}$	8	$1\frac{7}{8}$	20	8	$4\frac{3}{16}$	9	4
10	6	$2\frac{3}{8}$	8	$2\frac{1}{8}$	21	8	$4\frac{3}{8}$	9	$4\frac{3}{16}$
11	6	$2\frac{5}{8}$	8	$2\frac{5}{16}$	22	8	$4\frac{5}{8}$	9	$4\frac{3}{8}$
12	6	$2\frac{7}{8}$	8	$2\frac{1}{2}$	23	8	$4\frac{13}{16}$	9	$4\frac{9}{16}$
13	6	$3\frac{1}{8}$	8	$2\frac{3}{4}$	24	8	$5\frac{1}{16}$	9	$4\frac{13}{16}$

d_1 = diameter of screwed part of bolt $= \frac{7d + 1''}{9}$;

H = height of nut $= \frac{7}{8}d_1$ to $\frac{7}{8}d_1$.

When the bolts are $1\frac{1}{2}''$ diameter or over they are usually tapered, and tapered bolts are often made without heads. For taper of bolt use $\frac{3}{8}''$ in $12''$. (See Exercise 13.)

C = diameter of bolt centre $= D + 2.25d$.

While the shearing resistance of the bolts increases as the diameter C increases with the same diameter of bolt, yet to avoid the unnecessary use of material in the flanges, and secure a maximum of tightness in the coupling, the diameter C should be made as small as it is convenient to make it.

t = thickness of flange $= .3D$;

F = diameter of flange $= D + 3.9d$.

$e = D_2 - 1''$. $\qquad f = d_1$.

When cylindrical heads are used in tapered bolts their diameter may be $\frac{1}{4}''$ larger than the largest diameter of the bolt, and the height equal to $\frac{3}{4}d_1$.

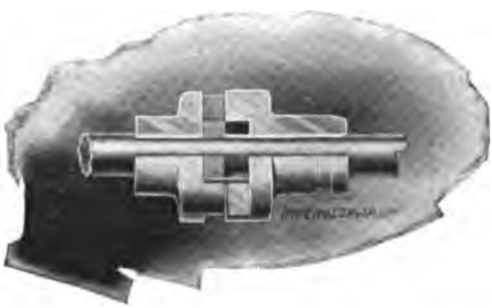


FIG. 126.

Jaw Clutch Coupling.—This coupling, shown in Figs. 126 and 127, is such that one half may be geared with or disgeared from the other half at will, and for slow-moving shafts this arrangement is simple and effective.

Exercise 51.—Make drawings as shown in Fig. 127 for a steel shaft 3″ diameter. The other dimensions may be taken from Table 23.

Spiral-jaw Coupling. This form of coupling, shown in Figs. 128 and 129, has axial engagement and is the commonest

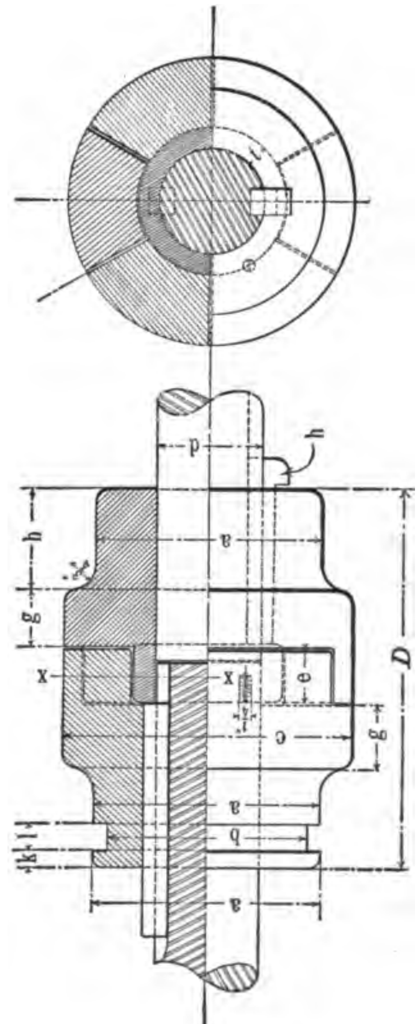


FIG. 127.

TABLE 23.

JAW CLUTCH COUPLINGS.

(Dimensions are in inches.)

d	*a*	*b*	*c*	*e*	*D*	*g*	*h*	*t*	*k*	*l*
15/16	2 3/4	2 1/4	3 5/8	3/4	5 1/2	1/2	1 3/4	1/4	1/4	1/2
1 3/16	3 1/4	2 3/4	4	7/8	6 1/4	5/8	1 7/8	1/4	3/8	1/2
1 5/16	3 1/2	3	4 1/4	15/16	6 3/4	3/4	1 7/8	5/16	3/8	1/2
1 7/16	3 3/4	3 1/8	4 5/8	1 1/16	7 1/8	7/8	2	5/16	3/8	1/2
1 11/16	4 1/4	3 5/8	5 1/4	1 1/8	7 1/2	1	2	3/8	7/16	1/2
1 15/16	4 3/4	4	5 7/8	1 3/8	8 3/8	1 1/8	2 1/8	3/8	7/16	1/2
2 3/16	5 1/4	4 1/8	6 1/8	1 3/8	9	1 1/4	2 1/8	7/16	7/16	5/8
2 7/16	5 1/2	4 3/4	7	1 9/16	10 1/8	1 3/8	2 3/8	1/2	7/16	5/8
2 11/16	6	5 1/4	7 5/8	1 9/16	10 1/2	1 1/2	2 1/4	1/2	9/16	5/8
2 15/16	6 1/2	5 3/4	8 1/4	1 11/16	10 7/8	1 5/8	2 7/16	9/16	9/16	5/8
3 7/16	7 1/2	6 3/4	9 1/8	1 15/16	12	1 3/4	2 13/16	5/8	5/8	5/8
3 15/16	8 3/8	7 3/8	10 3/4	2 1/16	13	1 7/8	3 1/8	11/16	11/16	3/4
4 7/16	9 1/4	8 1/8	11 7/8	2 1/8	13 3/4	2	3 5/16	13/16	3/4	3/4
4 15/16	10 1/2	9 3/4	13 1/8	2 1/8	15 3/8	2 1/8	2 7/8	13/16	13/16	3/4
5 7/16	11 1/2	10 3/4	14 3/4	2 3/8	16 7/8	2 1/4	4 1/4	7/8	13/16	3/4
5 15/16	12 1/2	11 3/4	16	3	18 3/8	2 3/8	4 3/8	15/16	15/16	7/8
6 7/16	13 3/4	13	17 1/8	3 1/4	19 7/8	2 1/2	5		1 1/16	1

of its kind. It is readily thrown in and out of gear by means of a lever and fork working in a groove shown at the right of

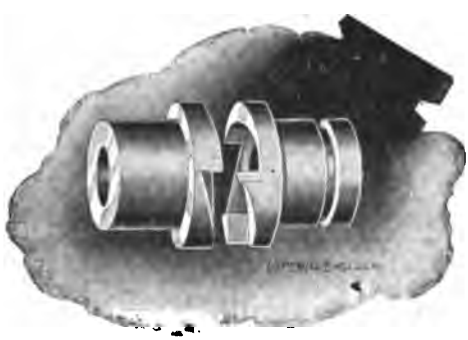


FIG. 128.

the figure. This style of clutch is adapted to the transmission of motion in one direction only.

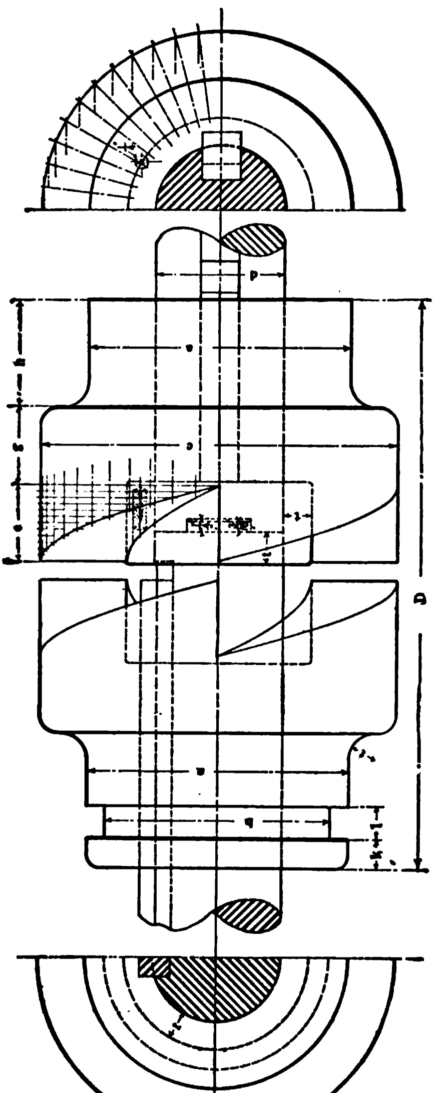


FIG. 129.

Exercise 52.—Make drawings of a spiral-jaw coupling for a $2\frac{1}{2}''$ shaft, as shown in Fig. 129. *Scale 6″ = 1 foot.*

The Universal-joint Coupling.—This most common of flexible couplings is best known as Hooke's coupling. Reuleaux says: "If not the original inventor of the Universal Joint, the Italian Cardan was the first to describe it (1501–1576), and the Englishman Hooke (1635–1702) first applied it for the transmission of rotary motion." The practical value of this form of coupling is that it can be used to connect two shafts whose axes intersect, and the angle between the shafts may be varied during rotation; this latter feature makes it suitable for ship propeller-shafts, to allow for the flexure due to the elasticity of the hull of the vessel.

The coupling shown in Fig. 130 is called a double-joint coupling because of the intermediate piece shown at S, and is such that two shafts in the *same plane* and making equal angles with the intermediate piece (S) will rotate with uniform angular velocity. This coupling is made by the Dodge Mfg. Co., Mishawaka, Indiana.

Exercise 53.—Make complete drawings as shown in Fig. 130. *Scale 6″ = 1 foot.*

Propeller-shaft Coupling.—Fig. 131 shows a propeller-shaft coupling, designed by the Campbell & Zell Co., Baltimore, Md. The material of the coupling and coupling-nut is wrought steel. The design is neat, compact, comparatively inexpensive, and has given good satisfaction.

Exercise 54.—Make drawings of Fig. 131 as shown, except that the diameter of the bolts is to be calculated by equations (20) and (21) (so that the *resistance to shearing* of bolts will be equal to *resistance to torsion* of shaft divided by R),

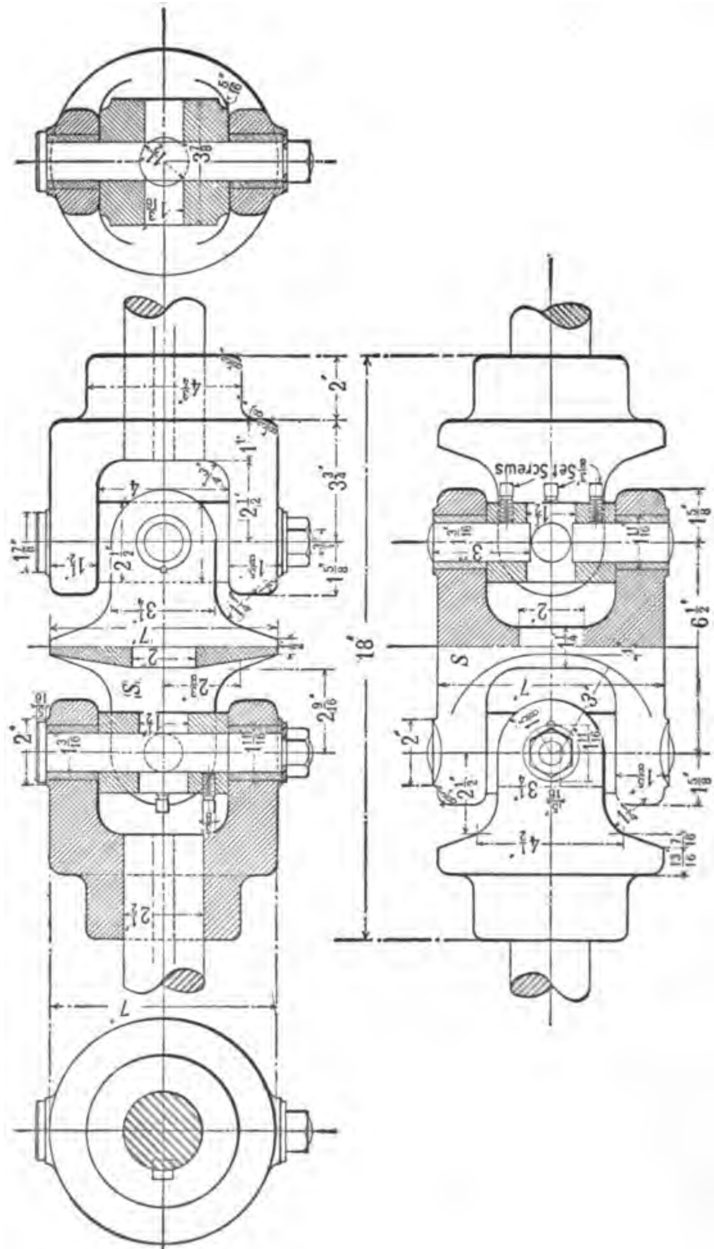
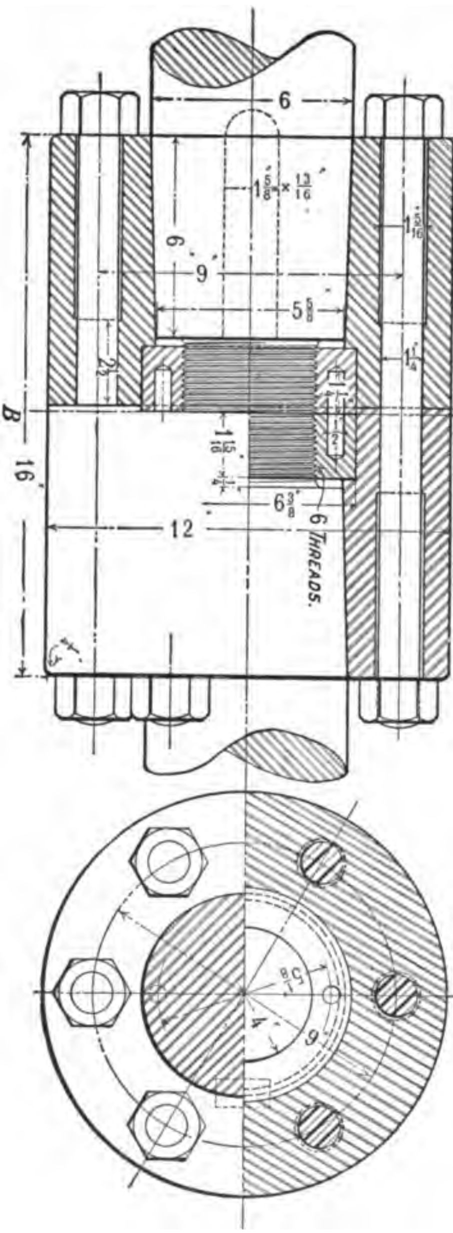


FIG. 130.



Section thro' A B

FIG. 131.

and make the diameter of the bolt-centre circle and the outer diameter of the coupling to suit. *Scale 6″ = 1 foot.*

In equation (20) use D equal to the mean diameter of the tapered part of the shaft, and from the result of equation (21) take the nearest commercial bolt diameter found in Table 8.

CHAPTER V.

PIPES AND PIPE-COUPLINGS.

Pipes.—Pipes are made of *cast iron*, *wrought iron*, *steel*, *copper*, and *brass*, and used to convey steam, water, or gas. Copper pipes are used most largely in marine work, and brass pipes or tubes are used to some extent in Europe for the fire-tube boilers of locomotives and for other purposes.

Thickness of Pipes to Resist Internal Pressure.

Let D = internal diameter of pipe or mean diameter for very thick pipes;

l = length of pipe in inches, inside of flanges;

P = internal pressure in pounds per square inch;

t = thickness of pipe in inches;

f_t = safe tensile stress in material in pounds per square inch.

Then the total force tending to separate two sections of the shell $= P \times D \times l$, which is resisted by the two thicknesses of the shell $\times$ the length of the pipe $\times$ the pressure per square inch, or $2(ltf_t)$; from this we get

$$t = \frac{PD}{2f_t}. \quad \ldots \ldots \ldots \quad (1)$$

This formula gives a thickness somewhat less than is used in practice.

D. A. Low gives

$$t = \frac{PD}{k} + c, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and the values of k and c as follows:

TABLE 24.

	k	c
Cast-iron steam or water pipes	4,000	0.3
Cast-iron steam-cylinders	3,500	0.5
Lap-welded wrought-iron tubes	17,000	0.06
Solid drawn steel tubes	40,000	....
Copper steam-pipes	7,000	0.1
Lead pipes	450	0.3

For foundry reasons cast-iron pipes should never be less than $\frac{5}{16}''$ thick, and long lengths not less than $\frac{7}{16}''$.

For tables giving the thickness of pipes for various pressures and equivalent heads see Kent's "Mechanical Engineers' Pocket-book," p. 189.

PIPE-COUPLINGS.

Cast-iron Pipe-couplings.—The most common method of connecting cast-iron pipes is by flanges cast on the pipes as shown in Fig. 131.

Exercise 55.—Make drawings of a cast-iron pipe-coupling like Fig. 131. $D = 8''$. Calculate remaining dimensions by the following formulæ. *Scale 6″ = 1 foot.*

$$t = 0.023D + 0.327;$$
$$F = 0.033D + 0.56;$$

$$E = 1.125D + 4.25;$$
$$C = 1.092D + 2.566;$$
$$d = 0.011D + 0.73;$$
$$n = \text{number of bolts} = 0.78D + 2.56;$$
$$w = \text{weight of pipe per foot} = 0.24D^2 + 3D;$$
$$W = \text{`` \quad `` flange} = .001D^3 + 0.1D^2 + D + 2.$$

This joint has the flanges faced all over, and is used for pressures up to 75 lbs. per square inch (170 ft. of head); for

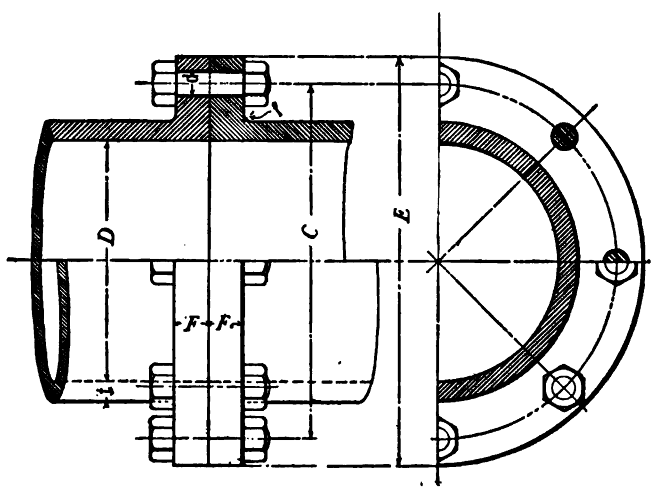


FIG. 131.

higher pressures the joint may be made with a string smeared with red lead between the flanges or a *lead*, *india-rubber*, or *gutta-percha* ring.

Exercise 56.—Make drawings of a cast-iron-pipe flange coupling, Fig. 132. Inside diameter of pipe to be 9″, other dimensions to be taken from Table 25. *Scale 6″ = 1 foot.*

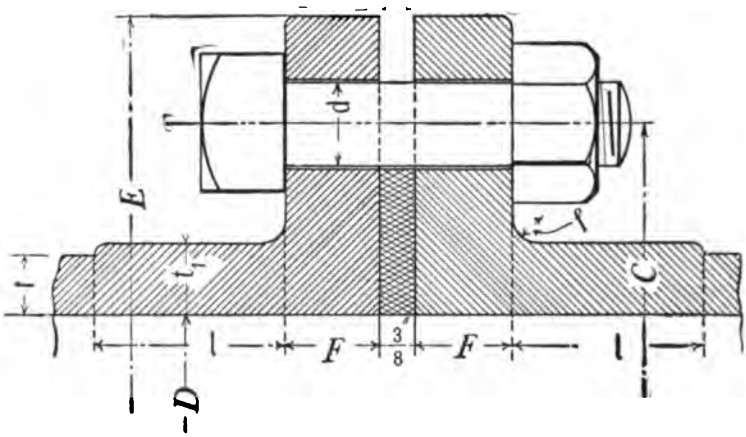


FIG. 132.

$l = 2F. \quad t_1 = t + .2t.$

TABLE 25.

STANDARD CAST-IRON FLANGES.

(Dimensions are in inches.)

D	*t*	*n*	*C*	*F*	*E*	*d*	Stress on Each Bolt per Sq. Inch at Bottom of Thread at 200 Lbs.	*f*	Stress on Pipe per Sq. Inch at 200 Lbs. Pressure.
2	.409	4	4¾	5/8	6	5/8	825	1/4	460
2½	.429	4	5¼	11/16	7	5/8	1050	1/4	550
3	.448	4	6	3/4	7½	5/8	1330	1/4	690
3½	.466	4	6½	13/16	8½	5/8	2530	5/16	700
4	.486	4	7½	15/16	9	3/4	2100	5/16	800
4½	.498	8	7¾	15/16	9½	3/4	1430	5/16	900
5	.525	8	8½	15/16	10	3/4	1630	3/8	1000
6	.563	8	9½	1	11	3/4	2360	3/8	1060
7	.60	8	10¾	1 1/16	12½	3/4	3200	3/8	1120
8	639	8	11¾	1⅛	13½	3/4	4190	3/8	1280
9	.678	12	13	1⅛	15	3/4	3610	3/8	1310
10	.713	12	14¼	1 3/16	16	7/8	2970	3/8	1330
12	.79	12	16½	1¼	19	7/8	4280	3/8	1470

This table was adopted by a conference of committees of the A.S.M.E. and the Master Steam and Hot Water Fitters Association in July, 1894. Sizes up to 24" diameter are designed for 200 lbs. pressure per square inch or less.

Spigot-and-socket Joint.—This is the usual joint for pipes that have to be embedded in the earth for conveying water or gas. Fig. 143 shows a joint of this kind. About half of the space between the spigot and socket is first filled with rope gasket and into the remaining half is poured molten lead, which when it cools is calked tightly into the socket with a hammer and round-nosed tool.

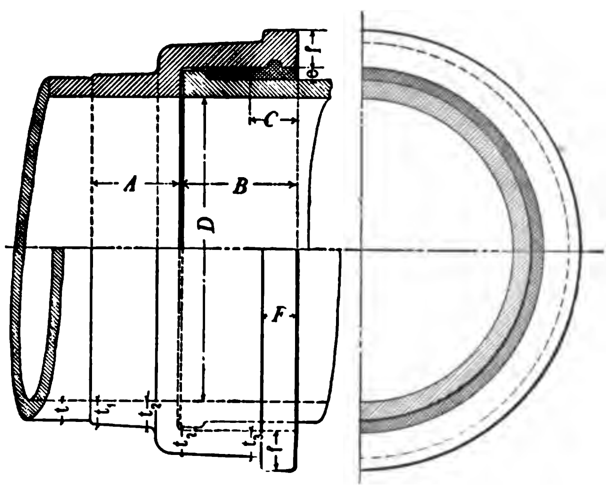


FIG. 133.

Exercise 56.—Make drawings of a spigot-and-socket coupling for an 8″ cast-iron pipe carrying a pressure of 100 lbs. per square inch (Fig. 133). *Scale 6″ = 1 foot.*

Same *elevations* and *sections* as in Ex. 55. Calculate the dimensions from the following proportions:

D = internal diameter of pipe;

t = thickness of pipe $= \dfrac{PD}{k} + c$ from equation (2);

$$
\begin{aligned}
t_1 &= t + .2t; \\
t_2 &= t_1 + \tfrac{1}{8}''; \\
t_3 &= t_2 + \tfrac{1}{8}''; \\
A &= .075D + 2\tfrac{1}{4}''; \\
B &= .1D + 2\tfrac{3}{4}''; \\
C &= .06D + 1''; \\
e &= \tfrac{5}{16}'' \text{ to } \tfrac{5}{8}''; \\
f &= .045D + .8; \\
F &= .04D + .7
\end{aligned}
$$

Exercise 57.—Make working drawings for the spigot-and-socket cast-iron pipe-coupling shown in Fig. 134. Internal diameter of pipe 10″. *Elevations* and *sections* similar to Ex. 55. *Scale 6″ = 1 foot.*

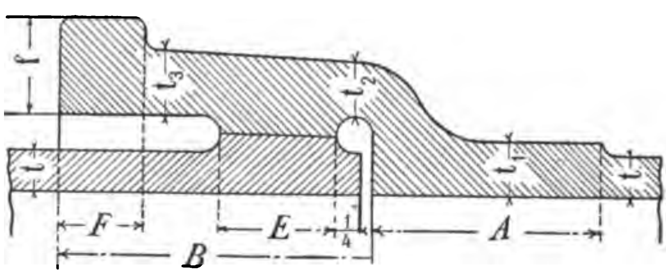


Fig. 134.

The dimensions for this problem are to be calculated from the proportions given for Ex. 56. The turned and fitted part E is made with a taper of $\frac{3}{8}''$ in 12″.

Exercise 58.—Make working drawings of an 8″ cast-iron-pipe flange coupling like Fig. 135. *Elevations* and *sections* as in Ex. 55. *Scale 6″ = 1 foot.*

Dimensions to be taken from Table 25.

These pipe-ends and flanges are strengthened with ribs drawn at an angle of 45° with the axis of the pipe, and the

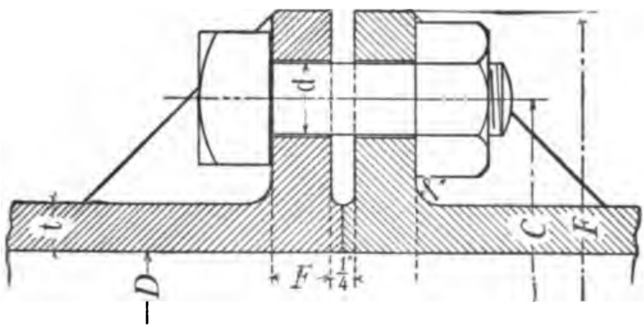


FIG. 135.

joint is made by means of *fitting-strips* cast on the flanges equal in width to the thickness of the pipe. The faces of these strips are finished perfectly square with the axes of the pipes, and before bolting up are smeared with red lead.

Exercise 59.—Make drawings of the *loose* flange coupling for a copper pipe shown in Fig. 136. Inside diameter of pipe 8″. *Scale 6″ = 1 foot.*

This joint is the invention of Mr. R. B. Pope of Dumbar-

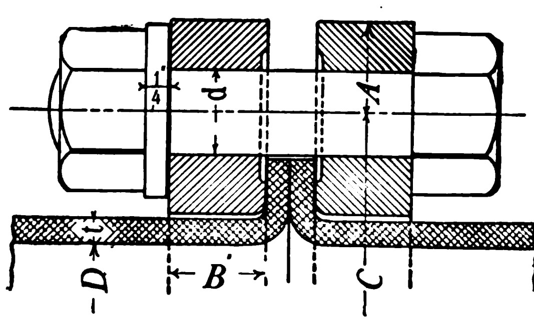


FIG. 136.

ton, Scotland, and is given by Low and Bevis. The flange rings may be made of cast iron, wrought iron, wrought steel, or cast steel; the latter is preferred. It is evident from Fig. 136 that the rings must be placed on the pipes before the ends are flanged.

These joints have been used for steam-, feed-, and exhaust-pipes from 1½″ to 36″ diameter.

The dimensions may be taken from the following table:

TABLE 26.

POPE'S PIPE COUPLINGS.

(Dimensions are in inches.)

D	*A*	*B*	*C*	*d*	No. of Bolts.	*D*	*A*	*B*	*C*	*d*	No. of Bolts.
1½	7/8	7/8	4	3/4	5	4½	15/16	1 1/16	7⅜	7/8	8
2	7/8	15/16	4¾	3/4	5	5	1	1⅛	8	1	8
2½	7/8	15/16	5¼	3/4	6	6	1	1 3/16	9⅛	1	9
3	15/16	1	5¾	7/8	6	7	1 1/16	1¼	10¼	1⅛	9
3½	15/16	1	6¼	7/8	6	8	1 1/16	1	11¼	1⅛	10
4	15/16	1 1/16	6⅞	7/8	7	9	1 1/16	1 5/16	12¼	1⅛	10

Wrought-iron and Steel Pipe-couplings.—Fig. 137 shows a very efficient form of joint for wrought-iron pipes. The flanged ends of the pipes are countersunk into the cast flange rings, and the bolt-heads are also countersunk about ⅜ of an inch. Between the flanged ends of the pipes is placed a ring of lead ⅛″ thick and from ⅜″ to ⅝″ wide.

Exercise 60.—Make drawings of the joint shown in Fig. 137 for a 6″ wrought-iron pipe. *Scale full size.*

A should be made equal to 1.25*D*. *t* may be taken from Table 7. Remaining dimensions may be taken from Table 26.

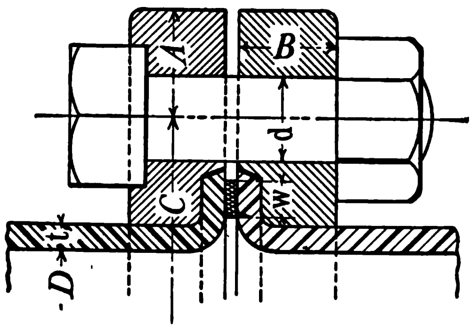


FIG. 137.

The "Converse" joint for wrought-iron and steel pipes is shown at Fig. 138. It is manufactured by the National Tube Works, McKeesport, Pa. This joint consists of a cast-iron sleeve with a space for lead at each end; there are also internal recesses plainly shown in Fig. 138, into which are

FIG. 138.

inserted rivet-heads on the ends of the pipes, and by a turn of the pipes the flanges become locked in position. Molten lead is poured into these recesses around the rivet-heads and tightly calked at the ends of the sleeves, as shown in Fig. 139.

Exercise 61.—Make drawings of the Converse joint for a 7″ wrought-iron pipe, according to the dimensions given in Fig. 139. *Elevations* and *cross-sections* same as in Ex. 55. *Scale 6″ = 1 foot.*

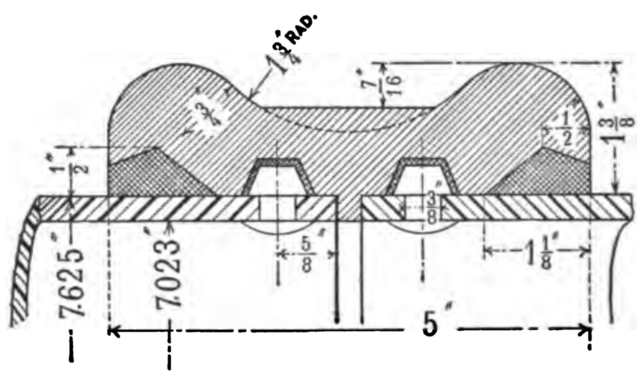


FIG. 139.

Screwed-flange Pipe-coupling.—Fig. 140 shows a wrought-iron pipe-joint made by screwing cast-iron flanges on the ends of the pipes and held together by bolts. It is used by the Philadelphia & Reading Coal and Iron Co. for their steam-pipes. The threads of the screws on the pipes are made according to the Briggs standard. The lugs shown in the figure on the right-hand flange are cast on, and have their inner surfaces finished to fit the cylindrical fitting-piece on the other flange. The ring shown between the flanges is of *gum rubber* and makes the joint steam-tight. The pipes are made in lengths of from 16 to 20 ft.

Exercise 62.—Draw a screwed-flange pipe-coupling like Fig. 140 for an 8″ wrought-iron pipe. *Scale 6″ = 1 foot.*

Dimensions may be taken from the following table:

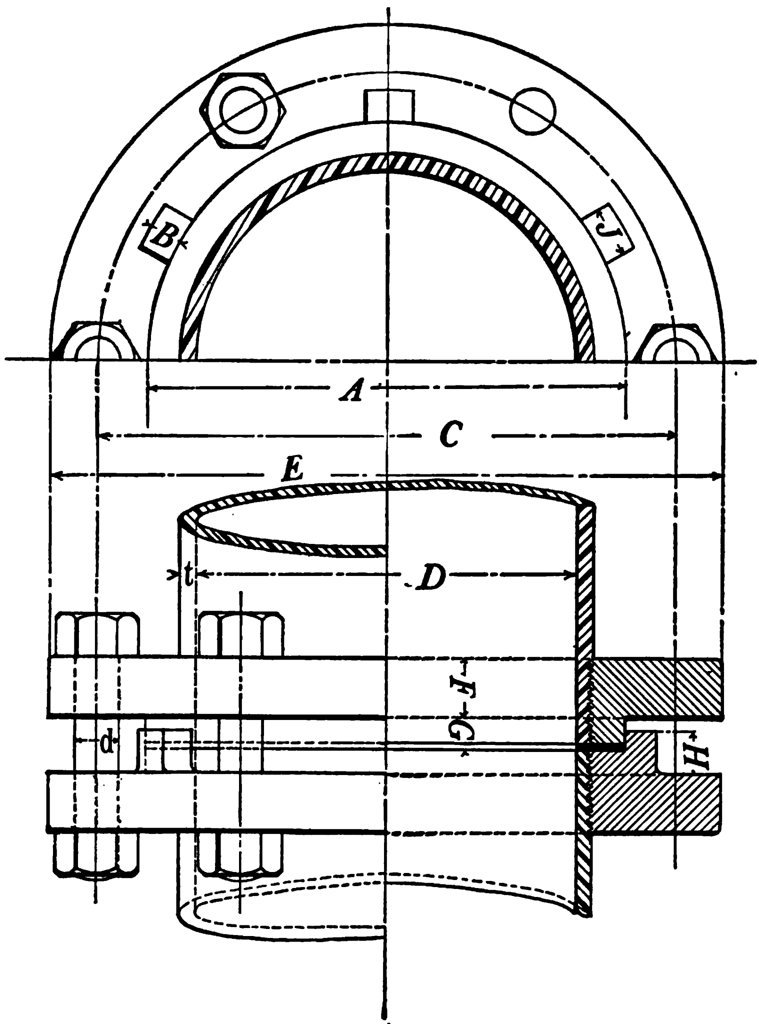


FIG. 140.

TABLE 27.

STEAM-PIPE CONNECTIONS OF PHILA. & READING COAL AND IRON CO.

(Dimensions are in inches.)

D	*f*	*A*	*B*	*C*	*E*	No. of Bolts.	*d*	*F*	*G*	*H*	No. of Lugs.	*J*
3	.217	5	1/2	6	7¾	4	3/4	3/4	1/2	7/8	4	3/4
3½	.226	5½	1/2	6¾	8½	4	3/4	3/4	1/2	7/8	4	3/4
4	.237	6	1/2	7½	9½	4	7/8	7/8	1/2	7/8	4	3/4
5	.259	7	1/2	8½	10½	4	7/8	7/8	1/2	7/8	4	3/4
6	.280	8	1/2	10	12	6	7/8	1	1/2	7/8	4	3/4
7	.301	9	1/2	11	13	6	7/8	1	1/2	7/8	4	3/4
8	.322	10	5/8	12	14	6	7/8	1¼	1/2	7/8	6	1
9	.322	11	5/8	13	15	8	1	1¼	1/2	7/8	6	1
10	.366	12	5/8	14	16¼	8	1	1¼	1/2	7/8	6	1

Screwed-socket Coupling.—Fig. 141 shows a screwed-socket coupling for a wrought-iron pipe. The socket is screwed half-way on to the end of one pipe, and the other pipe is then screwed into the remaining half of the socket. When it is not feasible to screw the long lengths of pipe into the projecting end of the socket, a screw is cut on one length of pipe and the socket is screwed fully on to this length, and when the pipes are butted together the socket is screwed back until it is half on each pipe.

For other wrought-iron or steel pipe-connections, see samples in drafting-rooms.

Exercise 63.—Make drawings of a wrought-iron socket pipe-coupling 7″ nominal diameter, to dimensions given in Fig. 141. *Elevations* and *sections* same as Ex. 55. *Scale full size.*

Locomotive Steam-pipe Ball Joint.—This joint (Fig. 142) is made between the steam *branch* pipe (*a*) and the tee-

pipe (*b*) which conducts the steam from the dome and dry-pipe to the steam-chests of the cylinders on each side of the engine. The pipes are of cast iron, and the spherical joint-ring

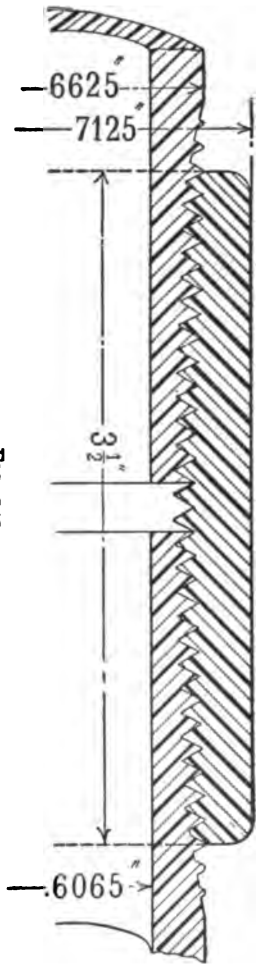


FIG. 141.

is of brass. The ball joint allows for expansion and contraction and for the pipe to be set at various angles with the perpendicular and horizontal.

Exercise 64.—Make drawings, as shown by Fig. 142, of a locomotive steam-pipe ball joint to dimensions given. *Scale 6″ = 1 foot.*

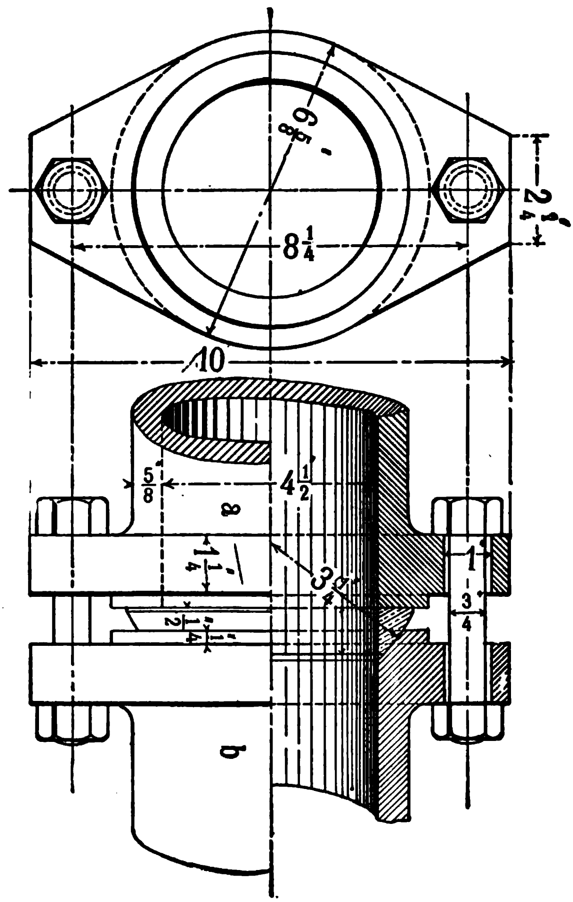


FIG. 142.

Wrought-iron Flange Pipe-coupling.—Fig. 143 shows a pipe-coupling made with angle-iron for a steel pipe. The angle-iron is rolled and welded into rings and riveted to the pipes. These flanges are used for either wrought-iron or

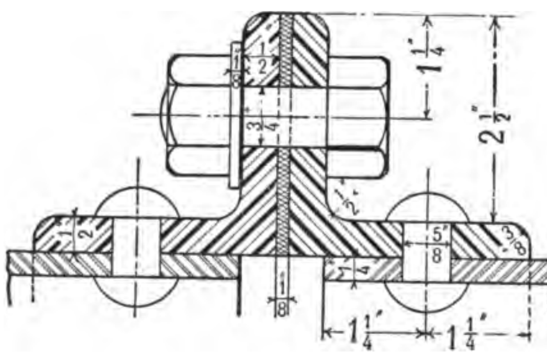


FIG. 143.

steel pipes. The joint is made steam-tight by means of a lead ring inserted between the flanges as shown.

Exercise 65.—Make drawings of a steel pipe with wrought iron flange coupling like Fig. 143. Nominal size of pipe 8″ diameter. *Elevations* and *sections* like Ex. 55. *Scale 6″ = 1 foot.*

Couplings for Brass and Copper Pipes.—The coupling shown in Fig. 144 is used on locomotive-boiler feed-pipes, injector-pipes, etc. The sleeves (*a*) and (*b*) are braced to the pipes, and a thin copper gasket placed between the ends of the sleeves makes the joint thoroughly tight when screwed up with the fluted nut (*c*).

Exercise 66.—Make drawings, as shown in Fig. 144, of a brass pipe-coupling, outside diameter to be 2½″. *Scale full size.*

The dimensions may be taken from Table 28.

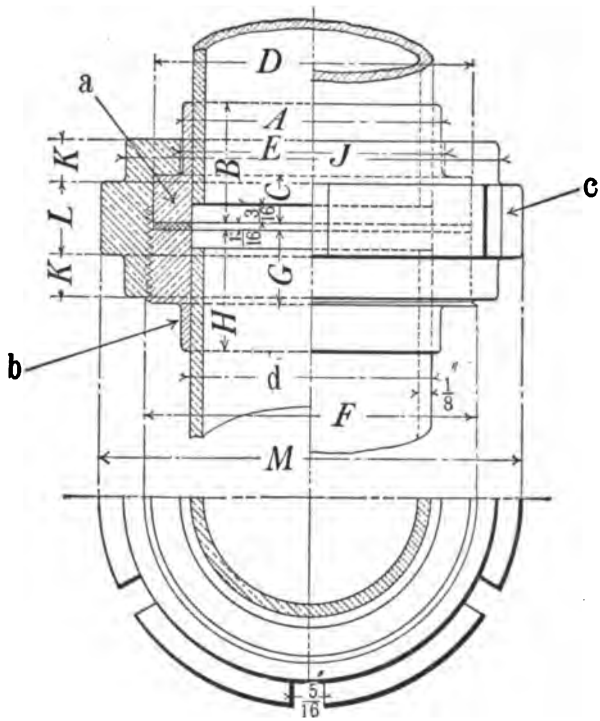


FIG. 144.

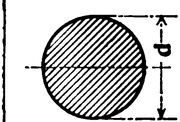
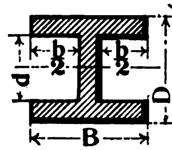
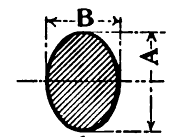
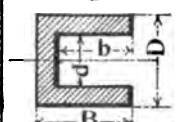
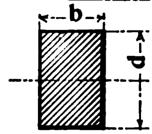
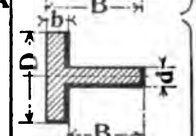
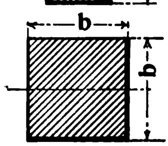
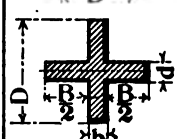
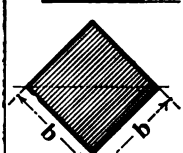
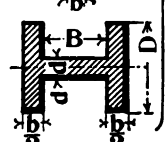
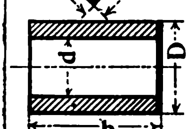
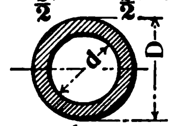
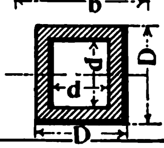
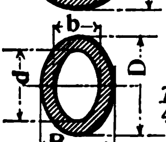
TABLE 28.

COUPLINGS FOR BRASS, COPPER, AND WROUGHT-IRON PIPES.

(Dimensions are in inches.)

d	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*t*
1 1/4	1 7/16	1	5/16	1 13/16	1 1/2	2	3/4	1	2 1/4	3/8	5/8	2 5/8	1/8
1 1/2	1 11/16	1	3/8	2 1/16	1 3/4	2 1/4	3/4	1	2 1/2	3/8	5/8	2 15/16	
1 3/4	1 15/16	1	3/8	2 5/16	2	2 1/2	3/4	1	2 3/4	3/8	5/8	3 3/16	
2	2 3/16	1 1/4	3/8	2 13/16	2 1/4	3	7/8	1 1/4	3 3/8	7/16	3/4	3 7/8	
2 1/4	2 7/16	1 1/4	7/16	3 1/16	2 1/2	3 1/4	7/8	1 1/4	3 5/8	7/16	3/4	4 1/8	
2 1/2	2 11/16	1 1/4	1/2	3 5/16	2 3/4	3 1/2	7/8	1 1/4	3 7/8	7/16	3/4	4 3/8	

TABLE 29.

FORM OF SECTION	AREA OF SECTION	MODULUS OF SECTION Z.	FORM OF SECTION	AREA OF SECTION	MODULUS OF SECTION Z.
	$\frac{\pi}{4}d^2 = .7854d^2$	$\frac{\pi}{32}d^3 = .0982d^3$.		DB-d	$\frac{1}{6}\frac{BD^3-bd^3}{D}$
	$\frac{\pi}{4}BA = .7854AB$	$\frac{\pi}{32}BA^2 = .0982BA^2$			
	bd	$\frac{1}{6}bd^2$			
	b^2	$\frac{1}{6}b^3$		Db+Bd	$\frac{1}{6}\frac{bD^3+Bd^3}{D}$
	b^2	$\frac{\sqrt{2}b^3}{12} = .118b^3$			
	$b(D-d)$	$\frac{1}{6}\frac{b}{D}(D^3-d^3)$		$\frac{\pi}{4}(D^2-d^2)$	$\frac{\pi}{32}\left(\frac{D^4-d^4}{D}\right)$
	D^2-d^2	$\frac{1}{6}\frac{D^4-d^4}{D}$		$\frac{\pi}{4}(BD-bd)$	$\frac{\pi}{32}\left(\frac{BD^3-bd^3}{D}\right)$

CHAPTER VI.

BEARINGS, SOLE-PLATES, AND WALL BOX-FRAMES.

ALL pieces employed in the transmission of power, rotating about a geometrical axis, must be supported in such a manner as to allow free rotation. The supports receive the general name of bearings, the various types being designated according to the direction of the pressure acting upon them. When the pressure is perpendicular to the axis of the shaft they are journal-bearings, and when bearings of this type and the framework connected with them are independent parts of a machine, they are indiscriminately called Plummer Blocks, Pillow Blocks, or Pedestals.

When the pressure is parallel to the axis of the shaft and the shaft terminates at the bearing surface, Fig. 164, the bearing is a pivot-bearing. When this type of bearing is employed for supporting the weight of a vertical shaft, it is termed a step- or footstep-bearing. When the pressure is parallel to the axis of the shaft and the shaft is continued through the bearing, the latter is termed a collar-bearing.

When pivot- or collar-bearings are used on horizontal shafts they are called thrust-bearings.

Journals are the parts of the shafts or axles that revolve on the bearings. They are made cylindrical, conical, or spherical, of which the cylindrical is the most common form.

To limit the longitudinal motion of journals the shafts are turned down or have collars forged upon them to form shoulders which come in contact with the faces of the bearings upon which the journals revolve. When practicable the length of journals should be about one per cent greater than that of their bearings.

The Area of a Bearing is the width of the chord of the arc in contact with the journal, multiplied by the length of the bearing. This is sometimes called the projected area, because it is the area of the contact surface projected on to a plane perpendicular to the direction of the pressure. Thus the area of a cylindrical journal-bearing, Fig. 164, is $D \times L$. The area of a pivot-bearing, Fig. 164, is $\frac{\pi D^2}{4}$. The area of a collar-bearing is $\frac{\pi}{4}(D^2 - D_1^2)N$. Where D is the diameter of the shaft D_1 is the outside diameter of collars and N the number of collars.

Solid Journal-bearings.—The simplest form of journal-bearing is made by drilling a hole through the frame of the machine, and to provide sufficient bearing surface the length of the bearing is increased by casting projections, which are termed bosses, upon the frame, as in Fig. 132. In this form of bearing there is no provision for wear, and the shaft can be returned to its initial position only by renewing that part of the frame that carries the shaft, or, when the hole wears oval, reboring the bearing sufficiently to fit it with a cylindrical sleeve or bush, as in Fig. 133. Such a bearing may be provided with a bush or lined with soft metal, and can be restored to its original condition by renewing the bush or lining. The end movement of the shaft may be limited

by making the diameter of one of the journals less than the diameter of the shaft, thereby forming a shoulder which limits the end movement in one direction, and securing a separate collar to the shaft, by means of a set-screw or taper-pin, in such a position as to limit the end movement in the other direction, as shown in Fig. 145. Another method is to make the shaft of uniform section throughout its length, limiting its end motion by means of two separate collars which may be arranged in three different positions.

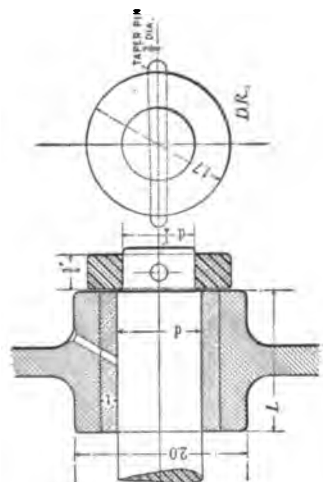


FIG. 146.

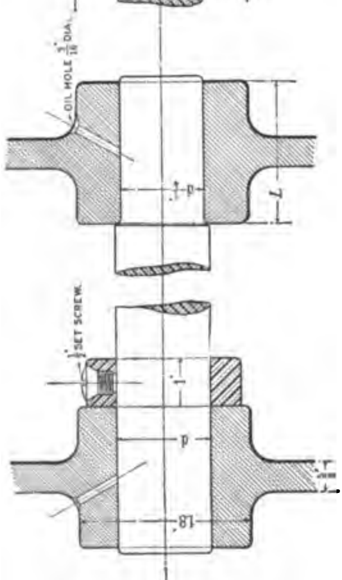


FIG. 145.

Exercise 67. — Draw two solid journal-bearings supporting a shaft 2″ in diameter, making the area of the bearing surface 6 square inches, and show an arrangement for limiting the end movement in either direction by means of one loose collar, as shown in Fig. 145. Draw also one bearing $1\frac{3}{4}$″ in diameter with a brass bush or sleeve, as shown in Fig. 146. Make t equal to $0.1d + \frac{1}{16}$″. Parts dimensioned in decimal

fractions are proportional to d. Complete and fill in the actual dimensions to the nearest sixteenth. *Scale full size.*

As the shafts supported by solid journal-bearings cast with the machine-frame have to pass through one bearing to the other, this form of bearing cannot be used when there are projections on the shaft. A solid bearing can be used, however, for supporting a shaft upon which there are projections, by making the bearings independent parts and securing them to the machine-frame by means of bolts. By this arrangement the shaft is turned down on the ends to form the journals, and one of the bearings is placed on its journal before it is secured to the frame. This form of bearing, Fig. 147, consists of a hollow cylinder cast upon a base through which bolts are passed into the machine-frame or supporting bracket.

Fig. 147 shows a design of a solid journal-bearing used for supporting the valve-gear reversing-shaft of a locomotive. Such a bearing can be used for this purpose because it is subjected to a comparatively light load, while the journal has a slow and intermittent movement. The length and shape of the bearing in this design are determined by local conditions, the bearing being carried forward further on one side of the base than on the other to suit the shaft. The width of the base is determined by the thickness of the frame, and is provided with strips on the under side to facilitate fitting.

Exercise 68.—Draw an **elevation** and **plan** of a solid journal-bearing of the form shown in Fig. 147, making $d = 2\frac{1}{2}''$ and $L = 2d$. The parts dimensioned in decimal fractions are proportional to d. *Scale full size.*

Construction.—First draw the centre lines and complete the cylindrical part of the bearing. Make the distance *a* equal to the outside radius of the cylindrical part + *r*, the

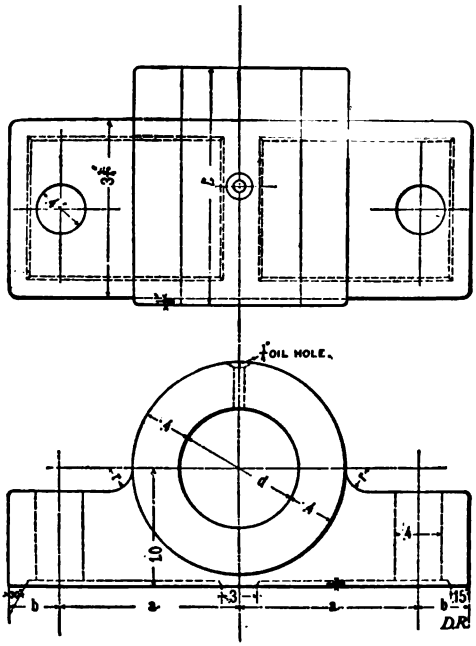


FIG. 147.

radius of the fillet, which we will make equal to, say, $\frac{1}{4}''$ + half the distance across the angles of the nut + $\frac{1}{8}''$ for clearance. The distance *b* can be made equal to half the distance across the angles of the nut + $\frac{1}{8}''$.

Divided Bearings.—Where the conditions are such that the shaft cannot be placed upon its bearings endwise, the bearings are parted and the parts fastened together by means of bolts or screws. The division is generally made on the line normal to the resultant pressures on the bearing.

In Fig. 148 is shown what is generally termed a two-part bearing. It consists of the block *P*, upon which the journal is supported, and the cap *C*, which is secured to the block by the bolts *CB*. In this design the journal is intended to be lubricated with semi-liquid grease which is passed through the opening *O*. The bearing is lined with Babbitt metal, $.08D + \frac{1}{16}''$ thick. The holes through which the holding-down bolts pass are made oblong to horizontally adjust the pedestal.

Wall Box-frames are built into the wall for the purpose of supporting a bearing for shafting which passes from one room or building to another. Fig. 149 shows a wall box-frame with an arched top to support the wall above it. On

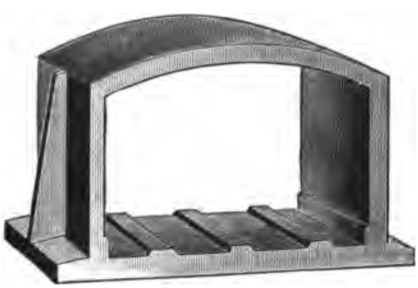


FIG. 149.

the sides are cast projecting webs *W* which fit into the wall to keep the frame from moving endwise. The upper side of the base is provided with raised machined strips *FS* upon which the pedestal rests, as shown in Fig. 150, and at each end of this surface are projections *S*, on the sides of the frame, which are also machined. To adjust the pedestal horizontally, wooden keys of the necessary thickness are fitted between the surface *S* and the pedestal base. The height *H* is equal

to the highest point of the pedestal cap when raised clear of the cap-bolts CB + about 6″ to allow the engineer to remove the cap. The length l_1 is equal to l, the length of the base, + the amount of horizontal adjustment allowed on the pedestal

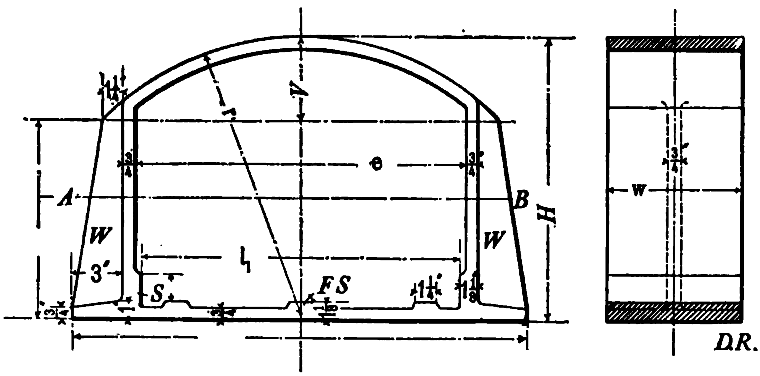


FIG. 150.

+ ½″. The width w is made to suit the thickness of the wall, which is usually built to average from 8″ to 12″. The proportioning of such a piece is largely a matter of experience, none of the parts being calculated for strength.

Exercise 69.—Draw a pedestal and wall box-frame of the designs shown in Figs. 148 and 150, placing the pedestal in position on the wall box-frame, to which it is secured by two square-headed bolts the heads of which project below the base. Make the pedestal to suit a shaft 2½″ in diameter, the length L equal to $3D$, and the width w of the frame equal to 8″. Show a **half-elevation** and **half-sectional elevation** of the pedestal, and an **elevation** of the wall box-frame, also a **plan** view of the pedestal with half of the cap removed, and in combination with this view show a section of the wall box-frame at the line AB. Make also an **end view** of the pedestal

and a **sectional end view** of the wall box-frame. All parts of the pedestal are proportional to the diameter D of the journal. Fill in all dimensions omitted. *Scale full size.*

Construction.—Draw the vertical and horizontal centre lines of the journal, then determine the distances from centre to centre of the bolts by drawing the line 1 which represents the top of the cap-flange, and the arc 2, which represents the top of the cap at the centre of the bearing. The centres of the cap-bolts can now be determined by making the corners of the nuts from $\frac{1}{16}''$ to $\frac{1}{8}''$ clear of the fillet which joins the lines 1 and 2. It is obvious that the bolts may be brought nearer together by either increasing the thickness of the cap-flange or cutting out the curve 2 around the nut, but on small pedestals for line shafting this is unnecessary. The radius r is made equal to half the distance across the angles of the nut $+\frac{1}{4}''$ for finish. The distance from centre to centre of the holding-down bolts is equal to the distance $b +$ the horizontal adjustment (equal to the length of the hole — diameter of bolt) $+$ the diameter of the washer $+$ the radii of the fillets, which may be made equal to about $\frac{1}{4}''$. Determine the radius r of the arched top of the wall box-frame by making V, the versed sine of the arc, equal to $\frac{e}{4}$.

Half the elevation is sectioned, to show more clearly the method employed to keep the Babbitt lining from turning with the shaft, the form of head on the cap-bolts, and also that the diameter of the holes through which the cap-bolts pass is greater than the bolt diameter. The plan view is shown with the cover removed from one side of the bearing, to show the form of that part of the bearing through which

the shaft passes. The fitting-strips on the under side of the base are of the same proportions as in the previous exercise. When practicable it is usual to provide the piece to which the bearing is fastened with fitting-strips also, as in Fig. 150.

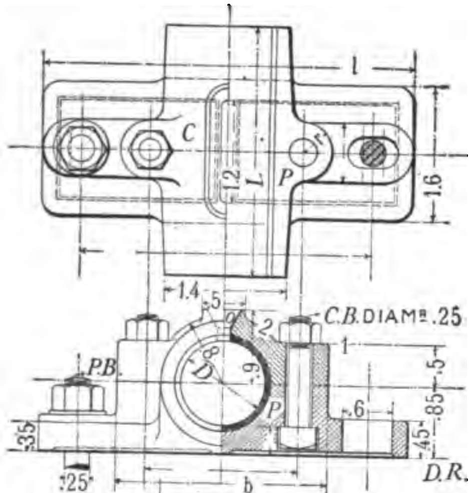
Post Bearings.—When the bearing has to be secured to a vertical surface, the base is cast on the side, as shown in Fig. 151. In the design shown in Fig. 152 it is necessary to provide the cap with four bolts because of the webs *W*, which are in the way of the bolts being placed on the centre as in Fig. 148. The bearing is arranged in this case for two grease-cups, which are screwed on to the cap at the tapped holes *O*. The cap-bolts are kept from turning when the nuts are being screwed down by projections *h* cast on the under side of the box.

Exercise 70.—Draw the **elevation** and an **end view half in section,** as shown in Fig. 152. Draw also a **plan view** of the top projected from the elevation. Make $D = 2\frac{3}{4}''$, and $L =$ three times D. Parts not dimensioned are in the same proportion to D as in the preceding exercise. *Scale half size.*

FIG. 151.

Construction.—Draw the centre lines of the bearing, taking care to leave sufficient space to draw the plan. Mark off the distance that the bearing projects from the post, then determine the length and width of the base. The centres of the bolts *PB* should be in a distance at least equal to the radius of the washer $+ \frac{1}{8}''$ from the ends of the base.

FIG. 148.



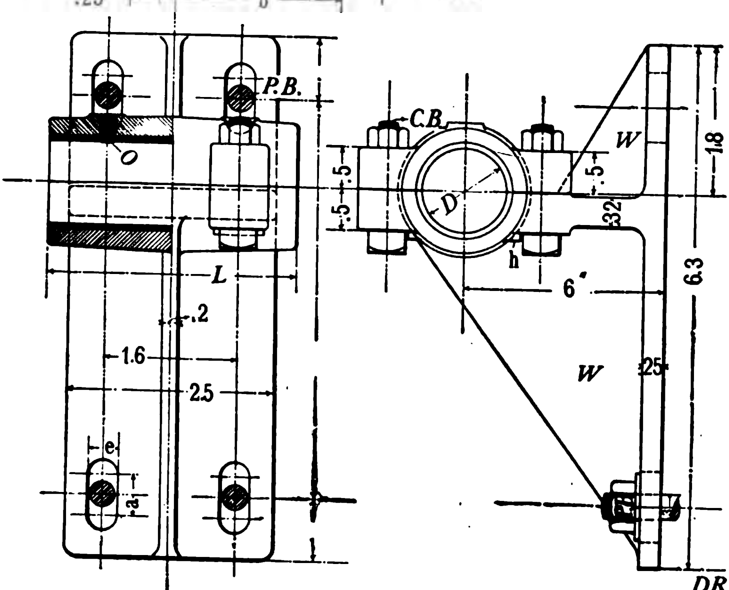


FIG. 152.

The vertical adjustment a is made equal to $1\frac{1}{4}''$. As the oblong holes are cored, the width e is $\frac{1}{8}''$ greater than the diameter of the bolts.

Wall Brackets are employed to carry pedestals which support a horizontal shaft running parallel and near to a wall. The bracket, Fig. 153, is fastened to the wall by means of three bolts which pass through it and the wall. The pedestal

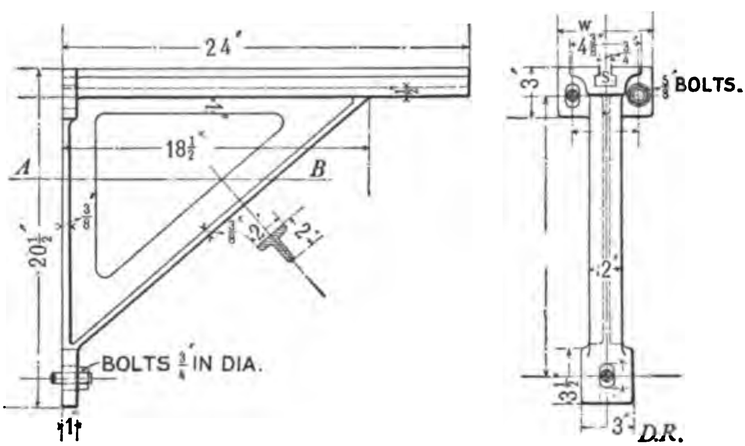


FIG. 153.

is secured to the upper surface by square- or T-headed bolts which slide in the ⊥-shaped slot S which runs the whole length of the bracket. By this arrangement the distance that the pedestal is from the wall can be adjusted.

Exercise 71.—Draw a wall bracket to the proportions given in Fig. 153. Make the slot S suitable for a $\frac{5}{8}''$ square-headed bolt. Draw also a **section**, the plane of section passing through the bracket at the line AB. *Scale half size.*

SELF-ADJUSTING BEARINGS.

Bearings for supporting line shafting may be divided into two classes, Rigid and Self-adjusting. When shafting is supported upon a number of rigid bearings it is essential that they all be in line, one with another, in order that the pressure be distributed over the entire surface of each. This is possible with bearings of the "rigid form" having comparatively long boxes when they are rigidly supported, but when supported upon insecure foundations, which are liable to sink, the bearing will assume such a position in relation to its journal as is shown in Fig. 154, where the entire load is carried

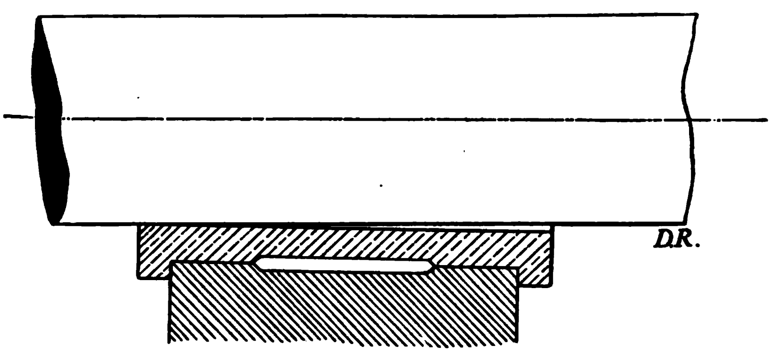


FIG. 154.

upon a small portion of the bearing. Such a condition exists also where the distance between the bearings is great in comparison with the shaft diameter, owing to the lateral deflection of the shaft by the gearing. Under such conditions the oil is forced out from between the rubbing surfaces, causing the metals to heat and seize by metallic contact.

To avoid this localization of pressure, bearings with a ball-and-socket joint are used, which to a limited extent adjust themselves to the various positions of the shaft, so that

the axis of the bearing will always coincide with that of the journal.

This form of bearing makes it practical to use a long box, thus keeping the pressure between the journal and bearing light enough to retain an unbroken film of lubricant between the rubbing surfaces. With these conditions the boxes may be made of cast-iron, which is the cheapest and, if well lubricated, the most desirable metal for the purpose. Many engineers, however, prefer to line these boxes with a white metal which rapidly wears and adjusts itself to any irregularities on the journal, making a perfect bearing more rapidly than would be the case with a harder material. Again, with the cast-iron box, should the lubricant fail and the metals come in contact, they will adhere and destroy the journal, while, under the same conditions, the babbitt metal would melt without materially injuring the shaft.

Drop Hanger-frame.—When a shaft is supported overhead and is not near a wall the bearings are carried upon a frame, called a hanger frame, which is secured to the ceiling girders. Two forms are used, the U form, which braces the bearing on both sides, as shown in Fig. 155, and the J, which braces one side only.

The objection to the U form as commonly made is the difficulty in getting the shafts in and out of the hanger. This has been overcome to some extent by making the hangers open at the bottom of the U, as it were, and connecting the sides with bolts.

The J form has the advantage of facilitating the mounting and dismounting of the shaft, but is liable to vibrate unless made comparatively heavy.

Fig. 156 shows a hanger made by the Dodge Manufacturing Co. which combines the advantages of both forms. This is attained by making the hanger open on one side and providing it with detachable links L, "which are split, and by bolts LB, drawn together upon taper cones C, cast on the

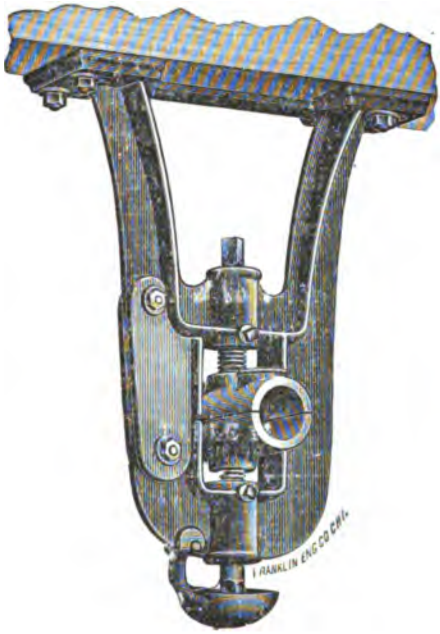


FIG. 155.

hanger frames F, which match corresponding recesses in the parts of the links. These links are thus drawn up to a positive bearing and form a connection which is virtually solid, and yet they are easily removed and replaced." Fig. 156 shows a shaft hanger with an adjustible bearing B, which is carried between the adjusting screws P and P', called the plungers. These plungers are screwed into the frame F and serve a double purpose; first, they are a means of obtaining a vertical adjust-

ment; second, they provide the sockets, with which the spherical surfaces on the box engage, to form the ball-and-socket joint. The plungers are locked in position by the set-screws *S*. The bearings are lubricated by filling the cups

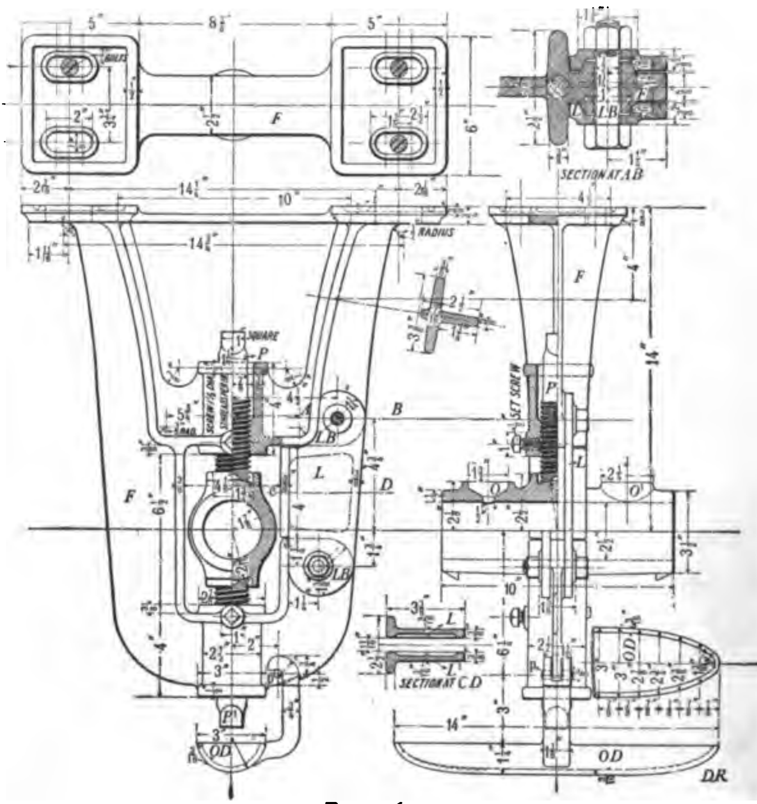


FIG. 156.

O and *O′* with grease, or cotton saturated with oil. The drippings of waste oil from the box are caught in the oil dish *OD* attached to the frame by hooking the head over the pin *P*, which is cast on the frame.

Exercise 72.—Draw the **front** and **end elevations** partly in

section, as shown in Fig. 156, a **half plan** and a **half-sectional plan** of the side to the right, the plane of section passing through the hanger at the centre line. *Scale half size.*

Draw also **full-size sections** of the frame, the plane of section passing through the hanger at the lines *AB*, *CD*, and *EF*.

Fig. 157 shows Sellers method of forming the ball-and-socket joint on adjustable hanger bearings. The plungers *P* and *P′* have shallow threads which extend along a portion of the plungers, while the threads in the boss are cut the entire length of the boss. The plungers are locked in position by the set-screws *S*, the points of which are made to press against the plain part of the plungers below the threads. The plungers are cast hollow, and are used as lubricators by filling them with cotton saturated with oil, which, under ordinary conditions, is sufficient to lubricate the journal. The openings *O* and *O′* are filled with tallow which is solid at ordinary temperatures but melts should the bearing become heated. The outer end of the plungers has a hexagonal hole to receive a key by means of which the screw is turned when adjusting the bearing.

Exercise 73.—Design a hanger-frame and bearing, altering the frame shown in Fig. 156 to suit the arrangement of plungers and bearing shown in Fig. 157, and design a method of fastening a drip-catcher to the frame, other than that shown in Figs. 155 and 156, which must be so arranged that it can be easily removed and replaced. Show a complete FRONT ELEVATION, SECTIONAL END VIEW, and PLAN projected from the front elevation.

Make $D = 2\frac{1}{8}''$, and length of bearing $= 4\ D$. *Unit of proportions is* $1.4\ D + .2$. *Scale half size.*

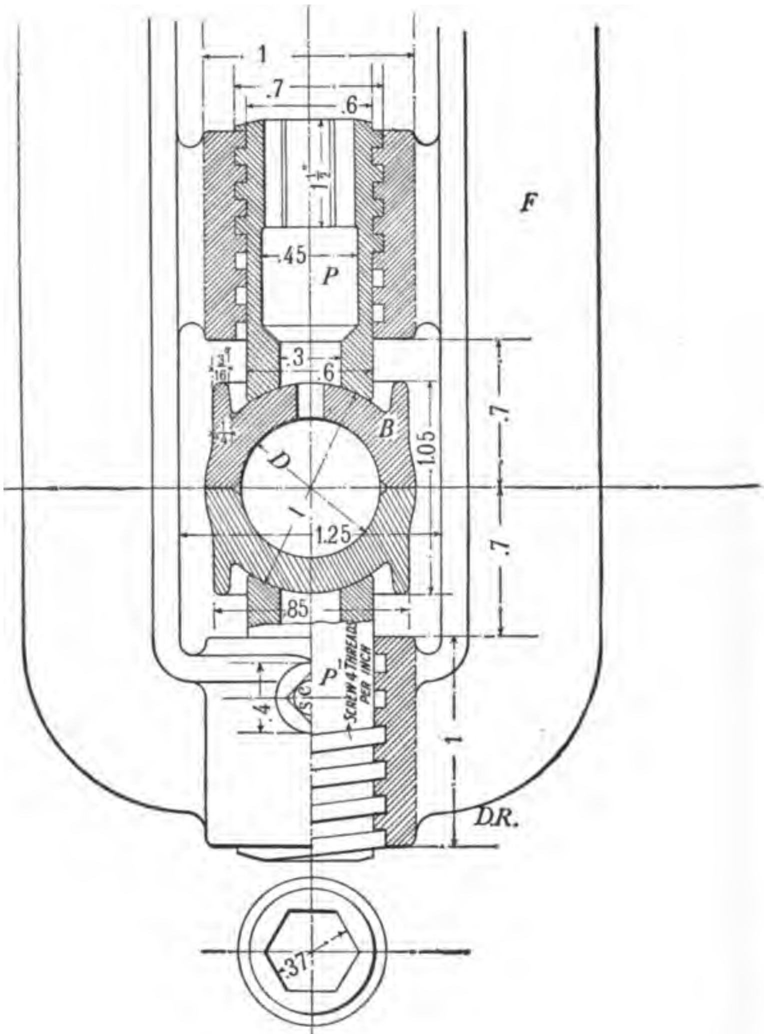

1
.7
.6
$1\frac{1}{2}$
.45
P
F
.3
.6
$\frac{3}{16}$
B
D
1.05
.7
.7
1.25
.85
P'
SCREW 4 THREADS PER INCH
.4
1
DR.
.37

FIG. 157

Wall- or Post-hanger is employed to serve the same purpose as the Wall bracket with its separate pedestal. The frames of these hangers are designed on the same

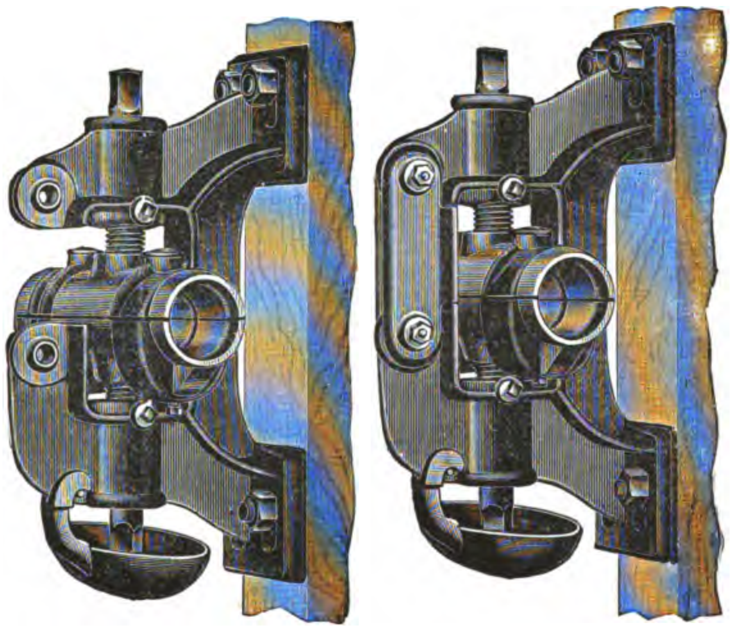


FIG. 158.

general lines and principles as the drop hanger-frames shown in Fig. 156. This hanger is shown in Fig. 158, with and without the double brace links, fitted with chain lubricating-bearings of the design shown in Fig. 160.

Exercise 74.—Draw FRONT ELEVATION and two END VIEWS as shown in Fig. 159, and a PLAN VIEW projected from the front elevation. *Scale 8″ to the foot.* Show also full-sized sections, the plane of section passing through the frame at the lines *AB*, *CD*, and *EF*.

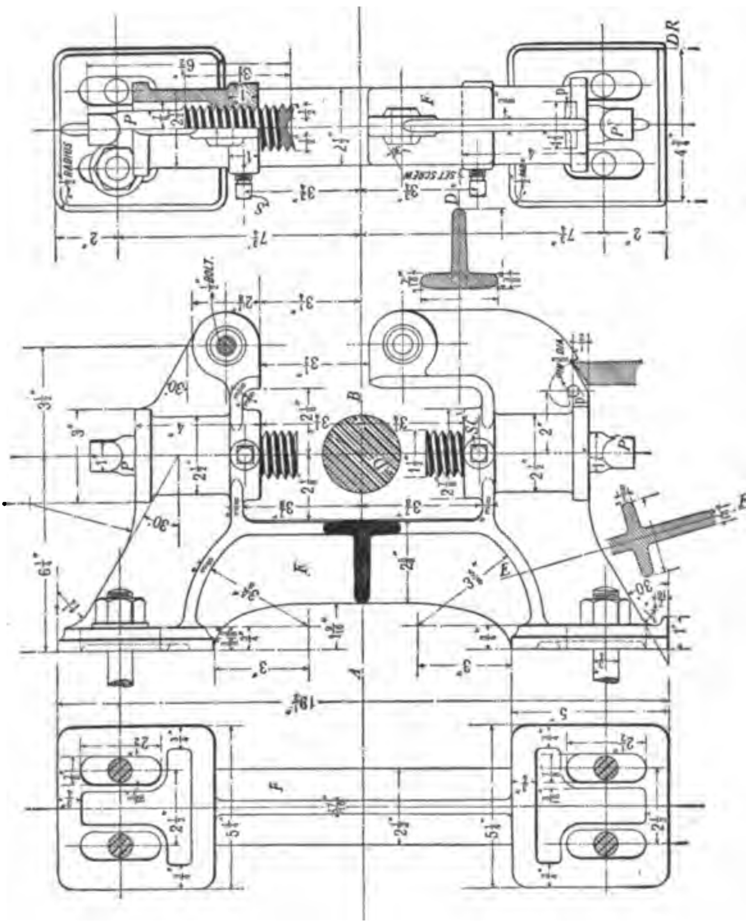


FIG. 159.

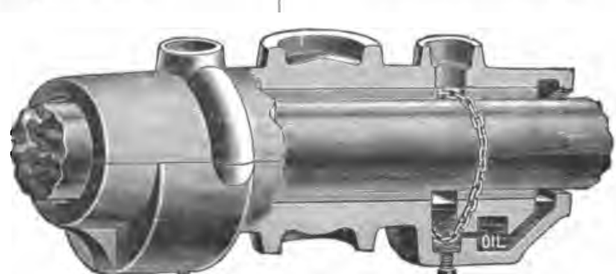


FIG. 160.

Chain Lubricating-bearing.—This type of bearing is designed to be lubricated by means of endless chains *C* which hang over the shaft, and as it revolves the chains revolve with it, passing through the oil in the reservoirs *O R* formed at each end of the box.

The chain *C* consists of a series of parallel links which form surfaces to which the oil adheres by capillary attraction, and is carried to the shaft, spreading through the channels *O C* to all parts of the bearing. All surplus oil falls back into the oil reservoirs, to be used again until it becomes thick or dirty, and is then drawn off by removing the plugs *S*.

Exercise 75.—Draw the chain lubricating-bearing shown in Fig. 161, showing a HALF ELEVATION and HALF SECTIONAL ELEVATION; an END VIEW projected from the right HALF SECTIONAL END VIEWS projected from the left-hand end, the plane of section passing through the bearing at the lines *AB* and *CD*, and a PLAN with half of the upper box removed. *Scale full size.* Draw also an ELEVATION AND PLAN of a part of the lubricating chain as shown in Fig. 163. *Scale four times full size.*

Construction.—Fig. 162 shows a method of finding the centres of the chain represented in the end-view in position on the shaft. In this construction the centres may be taken on the curve unless from the points 1 to 2, where the radius is small. At this part step off chords equal in length to the pitch of the chain, and, parallel to the chords, draw lines tangent to the arcs. The intersection of the tangent lines may be taken as the centres of the chain at that part.

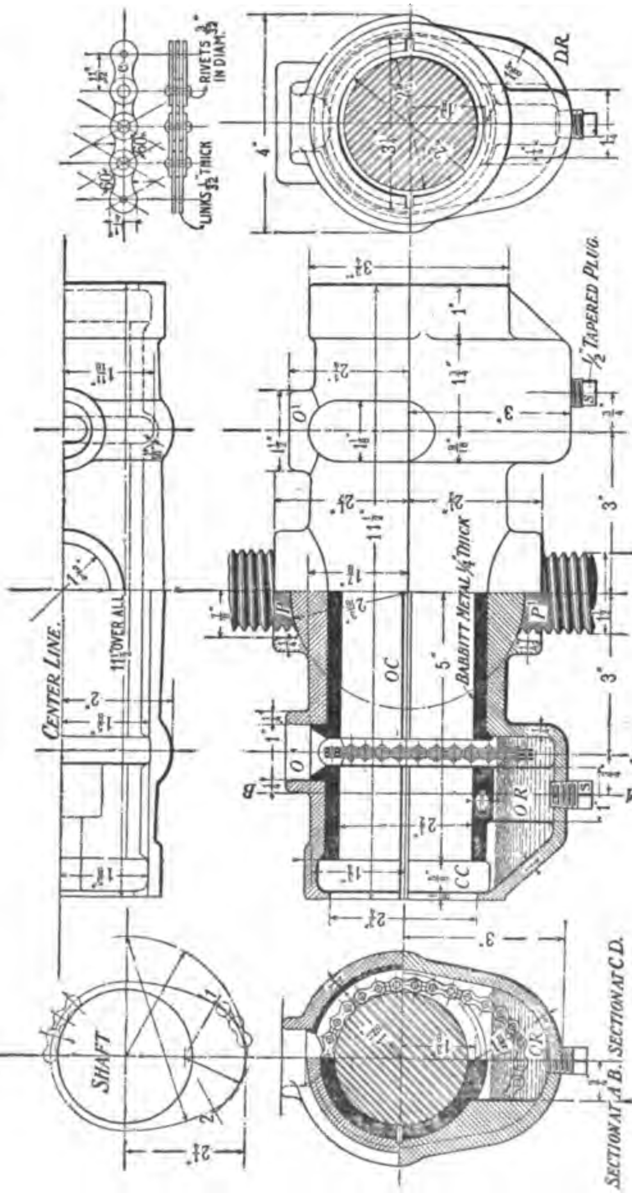


FIG. 163.

FIG. 162.

FIG. 161.

Bushes, Steps, or Brasses are names given indiscriminately to the bearings proper, i.e., the brass or bronze parts, that are in contact with and support the journal. They afford a means of taking up the lost motion due to wear, thus insuring that the journal with which they engage will have the required motion about the given axis. They must be made of a material that will allow the journal to run in contact with it with a minimum amount of friction, and will withstand wear without wearing the journal. They must also have sufficient strength to resist the stresses that come upon them, without undue yielding. When supporting a wrought-iron or steel shaft, gun-metal, to a limited extent, fulfils all these requirements. Other metals possess some of these qualities in a higher degree without having them all.

White metals, such as "babbitt's" or "magnolia" metals, offer less frictional resistance, and their surfaces may be destroyed without injuring the surface of the journal (as would be the case with the bronzes), but they are too soft to be used alone unless subjected to an exceptionally light load. The position of the bush in the supporting frame depends upon the direction of the pressure. In the majority of bearings the resultant pressures are in one or two directions, and all lost motion can be taken up by making the bearings in two parts. The ordinary forms of two-part bearings are shown n Figs. 164 to 167. The forms shown in Figs. 164 and 165 are turned, and the supporting frame is bored with a cylindrical hole into which the bearings are fitted. To prevent these forms from rotating with the shaft they are provided with rectangular lugs *L*, as in Fig. 165, or with steady pins *P*, as in Fig. 164.

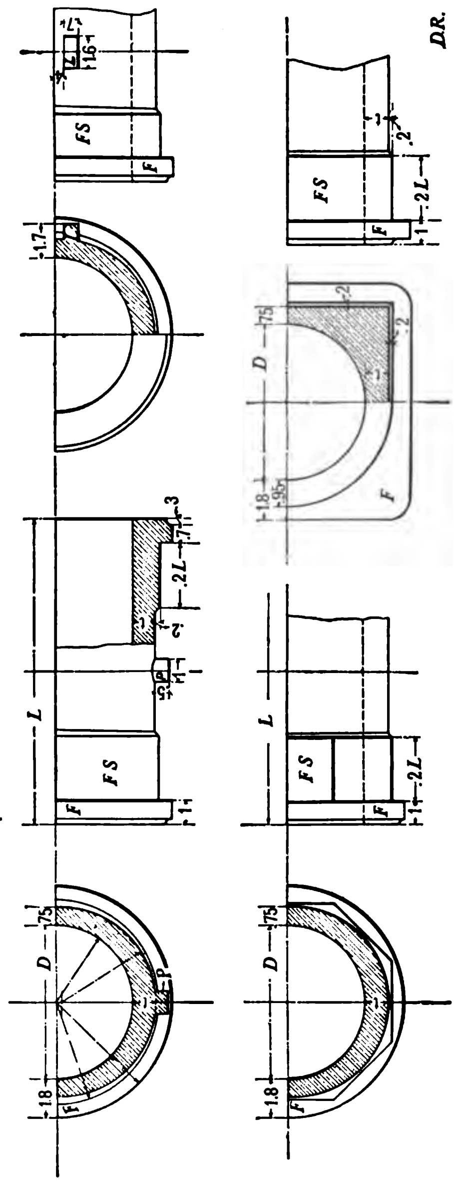


FIG. 164.

FIG. 165.

FIG. 166.

FIG. 167.

Unit $= t = .08D + .15$

The pins may be either cast with the bush or driven in. The forms shown in Figs. 166 and 167 are cast square or octagonal and planed to fit correspondingly shaped surfaces in the supporting frames. The square form is the cheaper, but should it become hot it is liable to be distorted, owing to the unequal distribution of metal. To facilitate fitting, and reduce machining on bearings, it is usual to support them at their ends only, by forming projecting faces *F S* at each end. This may be done successfully on small bearings subjected to a steady load, but on crank-shaft bearings it is advisable to support them over their length. The bearings should be divided on a line normal to the resultant pressures and, as they will wear very little at that part, they may be made thinner than at the part where the pressure is greatest. To keep the bearings from moving laterally along the shaft they are provided with flanges *F*, between which the supporting frame fits, as shown in Fig. 169.

Sole-plates.—When a pedestal is secured to masonry or brickwork it is necessary to spread the pressure upon the journal over a large surface. For this purpose a Sole- or Base-plate is employed. These usually consist of a flat cast-iron plate with a bevelled surface upon which the pedestal can be adjusted horizontally by means of the wood keys *K*, which are driven in between the joggles *J* and the ends of the pedestal base, as shown in Fig. 169. The pedestal is fastened to the sole-plate by the bolts *P B*, which pass through it and the base of the pedestal. The sole-plate is secured to the foundation by the bolts *F B*. The width (*b*) of the sole-plate should be equal to (*a*) width of pedestal base + the amount of movement of pedestal along shaft + say $\frac{1}{8}''$.

Adjustable Base-plates are used for adjusting bearings vertically and horizontally. The vertical adjustment is made by sliding wedges which may be arranged either laterally (as in Fig. 168) or longitudinally. The horizontal adjustment is

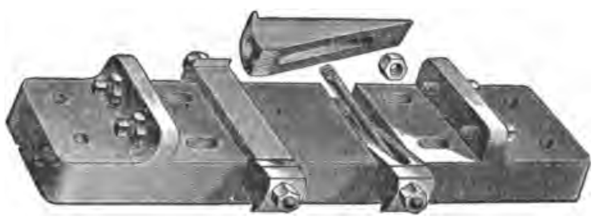


FIG. 168.

effected by means of set-screws which take the place of the wooden keys shown in Fig 169.

Pedestal or Pillow-block Bearings are used where it is necessary to have a bearing that is rigid and yet adjustable. Fig. 169 shows the ordinary form of pedestal bearing employed for supporting shafting from 3″ to 8″ in diameter. The inner surfaces of the block *P* and cap *C* are formed to suit the outer surface of the bushes. When the block is prepared by hand-work to receive the bushes it is provided with fitting strips *F S* to facilitate fitting, but when prepared by planing, the strips are unnecessary. Some engineers make the bushes that they do not touch each other when the shaft is in position, and as the bushes wear, a space being left between the cap and the pedestal, they are brought nearer together by screwing down the cap *C* by means of the bolts *C B*. To keep the cap from being screwed down too far, causing the bushes to bind the journal, the space between the cap and the pedestal is sometimes filled with hard wood and the wear is taken up by filing down the hard-wood distance-

pieces, thus allowing the cap to be screwed down a limited distance. Others make the bushes in contact with each other, as in Fig. 169, when the bushes fit the shaft, and when they become worn they are filed down sufficiently to compensate for the wear. When the bushes do not come in contact with each other and no distance-piece is used, the cap-bolts should be provided with double nuts. After the pedestal has been

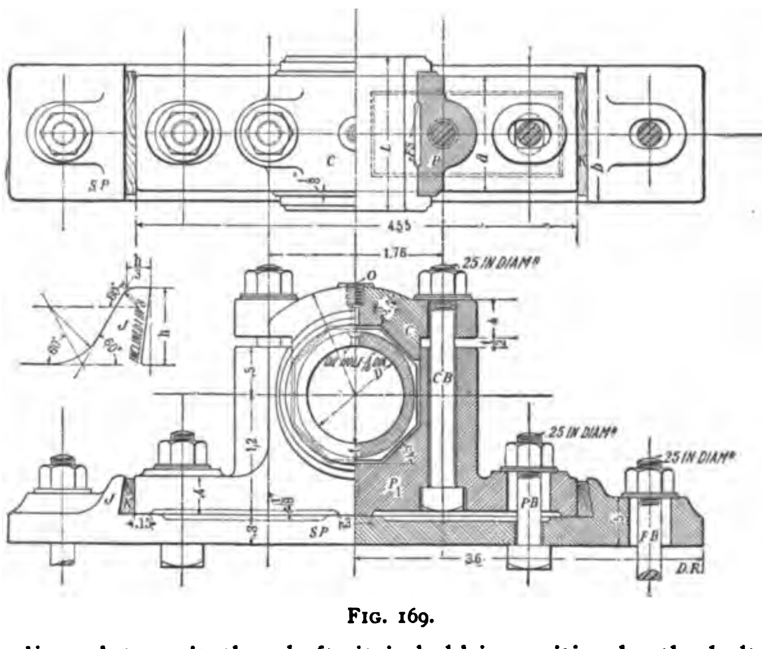


FIG. 169.

adjusted to suit the shaft, it is held in position by the bolts *PB*. The holes in the base- and sole-plate through which the bolts *PB* pass are made oblong to allow the pedestal to be moved along the shaft or transversely to it.

To facilitate the fitting of the pedestal to the piece upon which it is carried, the base is provided with fitting-strips around the edges and across the centre. The oil-cup is usually

cast with the cap C, or screwed into the tapped hole O, Fig. 169. On pedestals having journals less than 3″ in diameter O may be made to receive an oil-cup with a ¼″ pipe tap-shank, and when over 3″, with a ⅜″ pipe tap-shank.

Exercise 76.—Draw a general arrangement of a pedestal and sole-plate, Fig. 169, substituting the form of bearing shown in Fig. 164. Show a HALF ELEVATION and HALF SECTIONAL ELEVATION, the plane of section passing through the centre of the block; also a HALF PLAN and HALF SECTIONAL PLAN, the plane of section passing transversely through the centre of the journal. From the elevation project a HALF END-ELEVATION and HALF SECTIONAL END-ELEVATION, the plane of section passing through the centre of the pedestal. Make the length of the holes through the sole-plate and pedestal-base sufficient to allow the pedestal to move ¼″ in either direction. Make $D = 4''$ and $L = 2D$. *Scale half size.*

Construction.—All parts dimensioned in decimals are in terms of D (the diameter of the journal). Parts marked in inches are constant. Any parts not dimensioned can be determined by the student from knowledge derived from previous exercises. A method of drawing the joggles J is shown at Fig. 169, which will be readily understood from the drawing.

SELF-LUBRICATING PEDESTAL.

In this design, Fig. 170, an oil reservoir OR is formed on the under side of the bearing, in which loose rings R are revolved by their friction on the journal, thereby raising a continuous supply of oil to the upper side of the bearing, thus keeping the journal thoroughly lubricated and not

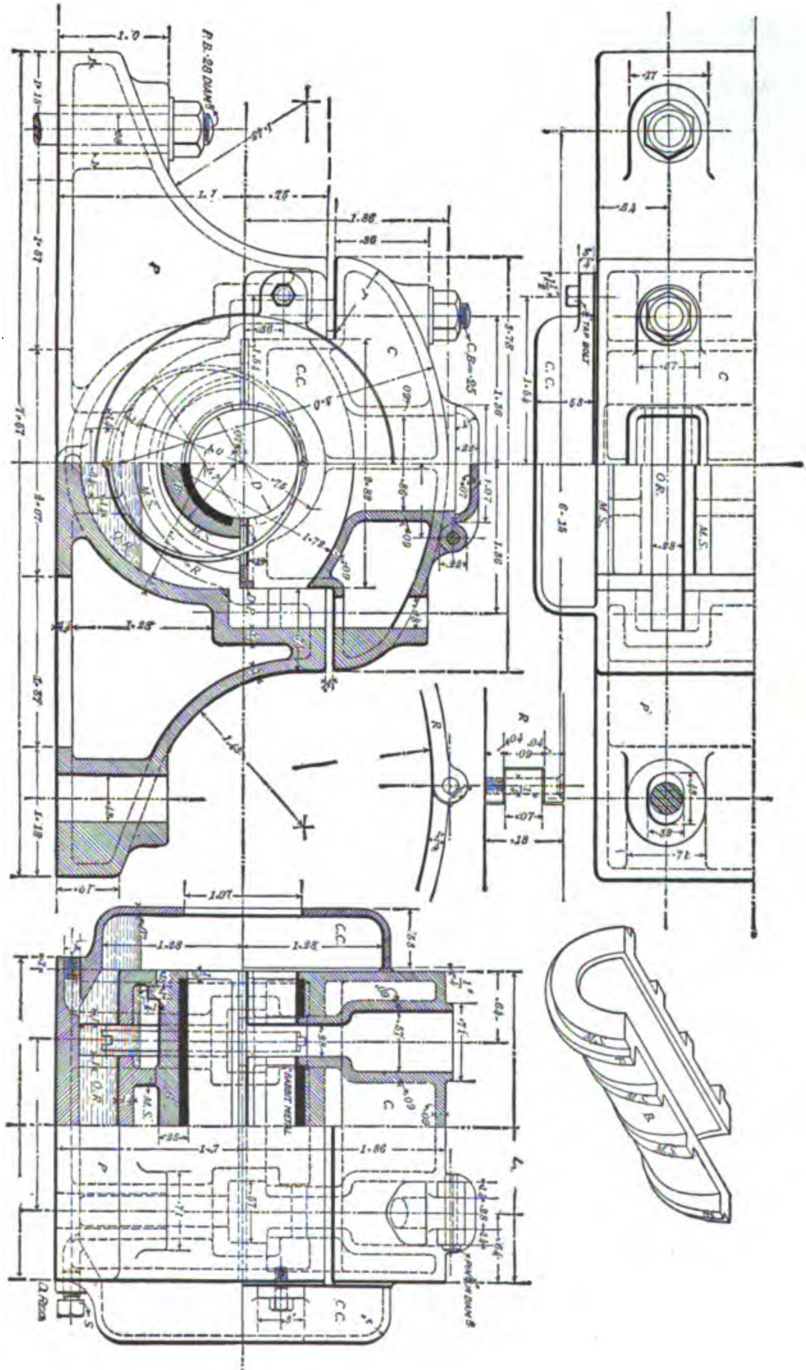


FIG. 170.

FIG. 171.

wasteful, as the surplus oil that flows out of the bearing is caught in the chambers *CC* and carried back to the reservoir *OR*.

As the same oil, in this form of lubricator, is being used repeatedly, after a time it becomes dirty and thick and is then useless. By removing the screws *S* the old oil is drained off, and the reservoir can then be replenished by pouring new oil into the openings in the cover. These openings are made large, so that the engineer can see if the rings are revolving.

This pedestal is designed for *down* pressure, and as there will be very little wear on the upper bush it is cast with the cap *C*. The lower bush *B* is a separate piece, as shown by the sketch, Fig. 171. To reduce the machining it is provided with projecting faces *MS*, called machining strips, which fit upon corresponding projections on the pedestal, and are made concentric with the shaft, so that to remove the bush it is not necessary to withdraw the shaft, as the bush when relieved from the load can be turned to the upper side of the journal.

By this arrangement the pedestal is practically independent of wear, as the bushes can be removed and re-babbitted with little trouble or expense.

To keep the bush from moving laterally, flanges *F* are cast at each end which fit inside of the end machining strips on the pedestal.

The lower bush is kept from turning by the distance piece *DP*, which also keeps the cap from being screwed down too far and clamping the shaft. To take up the wear of the bushes, the distance pieces *DP* are planed to let the cap go further into the pedestal. To allow this, a space

A is left between the pedestal and the cover. This space need not be greater than the thickness of the babbitt lining, which should be from $\frac{3}{8}''$ to $\frac{1}{2}''$ thick.

The cap is made to fit into the pedestal so as to sit squarely upon the journal, and does not depend upon the cap bolts to prevent lateral movement.

The cap is usually held down by two bolts, but to avoid large bolts in the larger sizes of pedestals it is quite common practice to use four. The bolts in this case are made square in section, and have T heads which fit into recesses cast in the pedestal. The pedestal is held in the proper position by the bolts *PB*, which pass through oblong holes in the pedestal to allow for longitudinal adjustment in either direction. This form of pedestal is suitable for journals from 5″ in diameter up.

Length of Bearings.—The frictional resistance at the surface of the journal converts the mechanical energy into heat, and, unless the area of the journal is sufficiently large to allow the heat to radiate as fast as it is generated, the temperature will become great enough to destroy the lubricant, allowing the rubbing surfaces to come in contact and adhere to each other. The radiating surface would be enlarged by increasing the diameter of the journal, but the velocity of the rubbing surfaces would also be increased; therefore the frictional resistance and the space through which it acts would be greater. Thus it will be seen that to add to the radiating surface without increasing the work at the surface of the journal we must increase the length of the bearing.

In a paper read before the Manchester (England) Associa-

tion of Engineers, Professor Goodman stated that the area of a bearing should be such that not more than one thermal unit of heat is generated per square inch of bearing surface per minute.

Let P = total pressure in pounds;

μ = coefficient of friction;

S = speed of circumference of journal in feet per minute = πDN;

N = number of revolutions per minute;

A = area of bearing, i.e., the diameter $D \times$ the length L;

D = diameter of journal;

L = length of journal.

Foot-pounds of work done per minute at the circumference of the journal = $P\mu S$. The thermal units per minute $= \frac{P\mu S}{778}$, and $A = \frac{P\mu SD}{778D}$, from which $L = \frac{P\mu S}{778D}$.

With steel journals running in bronze or white-metal bearings, having continuous lubrication, μ, the coefficient of friction may be taken at .0056.

Exercise 77.—Design a self-lubricating pedestal for a shaft 6″ in diameter, of the form shown in Fig. 170, to carry a load of 3500 pounds, and run at a speed of 300 revolutions per minute.

Show a HALF ELEVATION, HALF-SECTIONAL ELEVATION, the plane of section passing through the centre of one of the lubricators, a HALF END ELEVATION and HALF TRANSVERSE SECTION, the plane of section passing through the pedestal at the centre, a HALF PLAN of the left-hand side of the pedestal, a QUARTER PLAN with the cover (C) removed, a QUAR-

TER-SECTIONAL PLAN, the plane of section passing through the centre of the shaft. *Scale* 3″ *to the foot.*

Make also full-size drawings of the lower bush, showing a HALF ELEVATION and HALF-SECTIONAL ELEVATION, a HALF END VIEW, and a HALF TRANSVERSE SECTION, and a plan and elevation of the ring-joint as shown.

All points are proportional to the diameter (*D*) of the journal, except those parts which are constant for journals of various sizes.

CHAPTER VII.

BELT GEARING.

Belts.—Among the many different kinds of material used for belting are leather, cotton, gutta-percha, India-rubber, canvas, camel-hair, catgut, flat wire or hemp rope, steel bands, flat chains, etc.

The most common in general practice are leather and cotton, the latter often found coated with India-rubber and known as gum belts.

Leather is more durable than gum under most conditions, but for main driving the latter is superior, having an adhesion which is claimed to be one third greater than the former.

Transmission of Motion by Belts.—Motion may be transmitted from one pulley to another with uniform linear velocity by means of a belt, provided there is no slipping of the belt on the pulley; i.e., regarding the belt as inextensible every part of it will have the same velocity as the outside rim of the pulley.

Referring to Fig. 171, let d_1 and d_2 be the diameter of the driver and driven pulleys respectively, and let N_1 and N_2 be their revolutions per minute and V the velocity of the belt.

The speed of the rim of the driver

$$= d_1 \pi N_1 = V \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

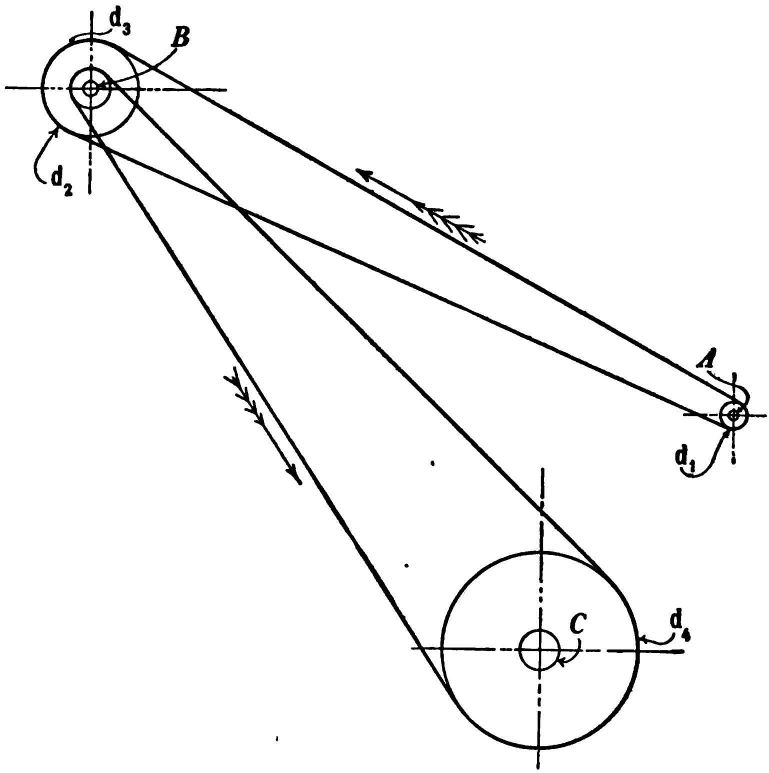


FIG. 171.

and the speed of the rim of the driven

$$= d_2 \pi N_2 = V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

therefore

$$d_1 \pi N_1 = d_2 \pi N_2 \quad \text{or} \quad d_1 N_1 = d_2 N_2 \quad \text{or} \quad \frac{N_1}{N_2} = \frac{d_2}{d_1}. \quad (3)$$

In all questions concerning the velocity ratio of belting the pulley diameters should be taken to the centre of the belt

thickness; thus the virtual diameter of the pulley would be the nominal diameter plus *one thickness* of the belt. For other calculations the thickness of the belt is so small it may be neglected without much error.

Example 1.—In the draughting-room at Sibley College there is a valve-motion model driven by an electric motor. The shaft A of the motor carries a pulley $1\frac{1}{2}''$ diameter from which passes a belt to a $15''$ pulley on a counter-shaft B. This shaft carries another pulley $6''$ in diameter connected by a belt to the driving-wheel pulley of $30''$ diameter on the valve-motion model axle.

The speed of the motor is 1450 R. P. M. Find the speed of the valve-motion model in R. P. M., Fig. 171.

From formula (3) we get

$$\frac{N_1}{N_4} = \frac{d_2}{d_1} \times \frac{d_4}{d_3} = \frac{15}{1.5} \times \frac{30}{6} = 50.$$

Substituting we get

$$\frac{1450}{N_4} = 50 \quad N_4 = \frac{1450}{50} = 29 \text{ R. P. M.}$$

Some Practical Rules.—The width of belts should be about 25 per cent less than the face of the pulley.

It has been demonstrated by experience that large pulleys and fast-running belts are much more economical than small pulleys and slow-speed belts. All pulleys should be carefully centred and balanced on the shaft. Driving-pulleys carrying shifting-belts should have a perfectly flat surface. All other pulleys should have a convexity of $\frac{1}{8}''$ to $12''$ of width; when curved the chord of the arc should be the same. For

pulleys smaller than 12″ wide, from ⅜″ to ½″ per foot of width should be used.

Pulley diameters should be as large as can be used provided the belt speed is kept within 5000 feet per minute, which is held to be the limit of speed for belt economy.

With regard to the position of idle pulleys in relation to the driving-pulley Taylor says, "Idle pulleys work most satisfactorily when located on the slack side of the belt about one quarter away from the driving-pulley."

Transmission of Power by Belts.—Let two pulleys A and B be connected by a belt with a tension equal to T_2. Until force is applied at A tending to produce rotation of the pulleys, the tension T_1 and T_2 will be equal; but as the force at A increases the tension in T_1 will increase, and that in T_2 will decrease until $T_1 - T_2 = P =$ resistance to rotation at the rim of the pulley; i.e., when the belt is at the point of slipping, the ratio of T_1 to T_2 will be a maximum and $= efa$, or $T_1 \div T_2 = efa$. Where e is the base of the Naperian system of logarithms, f is the coefficient of friction $= .3$, a is in π measure and $= a$ in degrees $\times$ 0.0174.

By logarithms we find that $T_1 \div T_2 = efa =$ log. $T_1 \div T_2 = fa$ log. $e = .4343fa$.

Example 2.—A six H.P. dynamo is to have a speed of 1450 R. P. M and has a 6″ pulley on its shaft. Power is obtained from an engine fly-wheel running at 58 revolutions per minute. To obtain the required velocity ratio between the engine and dynamo, the diameter of the fly-wheel will have to be 25 times that of the dynamo pulley with direct connection; but such a diameter would be practically impossible, so it will be necessary to install a counter-shaft. Let 18″ be the

most suitable diameter for the largest pulley on the counter-shaft, then the necessary speed of the counter-shaft will be $= 1450 \times \frac{6}{18} = 483$ R. P. M. Between the engine and counter-shaft the pulley diameter ratio $= \frac{483}{58} = 8.32$. Let the diameter of the fly-wheel be 50″ then its connecting-pulley on the counter-shaft will be $\frac{50}{8.32} = 6''$ nearly.

To determine the size of belt necessary to connect the dynamo with the counter-shaft we will have to find the value of $T_1 =$ to the working pull on the lower side of the belt.

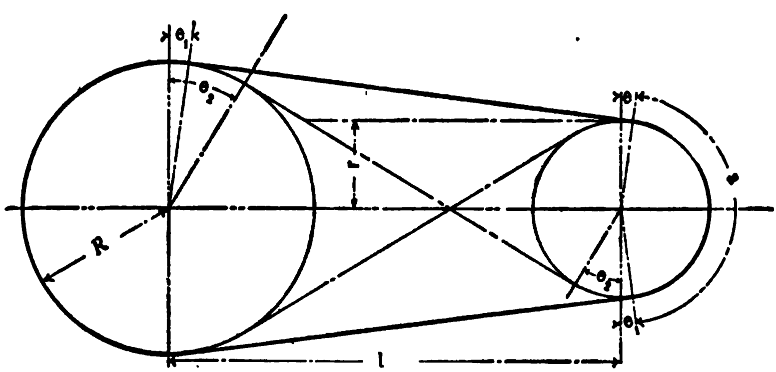


FIG. 172.

First find the work done by the dynamo $= 6 \times 33{,}000 =$ 198,000 foot-lbs. per minute; the rim of the dynamo pulley runs at $\frac{6\pi}{12} \times 1450 = 707$ feet per minute; therefore $T_1 - T_2 = \frac{198000}{707} = 280$ lbs. Let the centres of the dynamo shaft and counter-shaft be 15 feet apart, then (see Fig. 172)

tan. $\theta = \frac{R - r}{l} = \frac{9'' - 3''}{150} = .04$, and from a table of natural trig. functions we find that tan. $.04 = 2.25°$.
$a = 180° - 2\theta = 175.75$, a in π measure $= 175.75 \times 0.0174 = 3.05$. Then log. $T_1 \div T_2 = .4343 \times .3 \times 3.05 = .3974$; from a table of logarithms we find that .3974 is the log. of the number 2.50, therefore

$$T_1 \div T_2 = 2.50.$$

Combining these equations thus:

$$2.50\ T_1 - 2.50\ T_2 = 280 \text{ lbs.} \quad T_1 - 2.50\ T_2 = 0.$$

$1.50\ T_1 = 700$ we find $T_1 = 700 \div 1.50 = 466$, and allowing 70 lbs. per inch width of belt then

$$466 \div 70 = 6.66, \text{ say } 7'' \text{ nearly.}$$

Some Practical Rules for the Transmission of Power.—Richards gives the following rule for the size of driving-belts, which he says is near enough for all cases that arise in ordinary practice.

$$\text{H.P.} = \frac{V \times W}{A}. \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Where V = the velocity of the belt in feet per minute.
W = the width of the belt in feet.
A = the area given to suit different conditions in the following table:

TABLE 30.

LEATHER BELTS SINGLE THICKNESS.	1 H. P.	GUM BELTS AVERAGE THICKNESS.	1 H.P.
On smooth iron pulleys	80 ft.	On smooth iron pulleys	60 ft.
On wooden pulleys	65 ft.	On wooden pulleys	50 ft.
On covered pulleys	50 ft.	On covered pulleys	35 ft.

Belts should be made as wide as possible; they are often too narrow, but never too wide.

Thickness of Belts.—As belts increase in width their thickness should also increase. Double belts should be used on pulleys over 12″ diameter. Large belts running at very high speeds, as in electrical work, should have slots punched through them in such manner and position as to prevent air cushion.

The following proportions for thickness of belt and corresponding working tension, based on a safe working stress of 320 lbs. per sq. in. for laced joints, are given by Unwin:

TABLE 81.

Thickness of belt......	$\frac{3}{16}$″	$\frac{7}{32}$″	$\frac{1}{4}$″	$\frac{5}{16}$″	$\frac{3}{8}$″	$\frac{7}{16}$″	$\frac{1}{2}$″	$\frac{9}{16}$″	$\frac{5}{8}$″	$\frac{11}{16}$″	$\frac{3}{4}$″
Working tension in lbs. per inch of width....	60	70	80	100	120	140	160	180	200	220	240

For other rules and formulæ see Kent's Engineers' Pocket Book, page 876.

For a safe working tension under ordinary conditions, many authorities allow only 45 lbs. per inch of width; but according to Mr. A. W. Smith, experiments have shown that a safe tension of 70 lbs. may be had per inch of width of belt.

Proportions of Pulleys (Figs. 173 and 174).—

a = centre of set-screw from end of hub = $1\frac{1}{4}d_1$.

a_1 = centre of bolt from edge of flange = $1\frac{1}{4}d_2$.

b = width of belt—see Example 2.

B = pulley face = $\frac{9}{8}(b + 0.4)$. (Unwin) . . . . . (5)

d = shaft diameter.

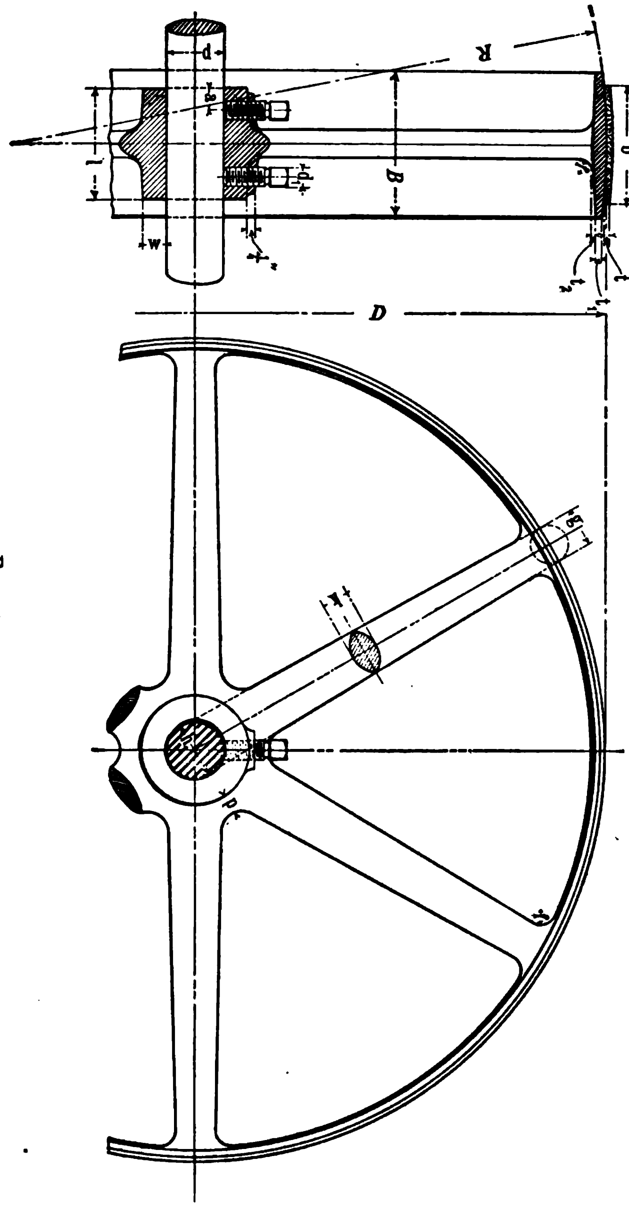


FIG. 173.

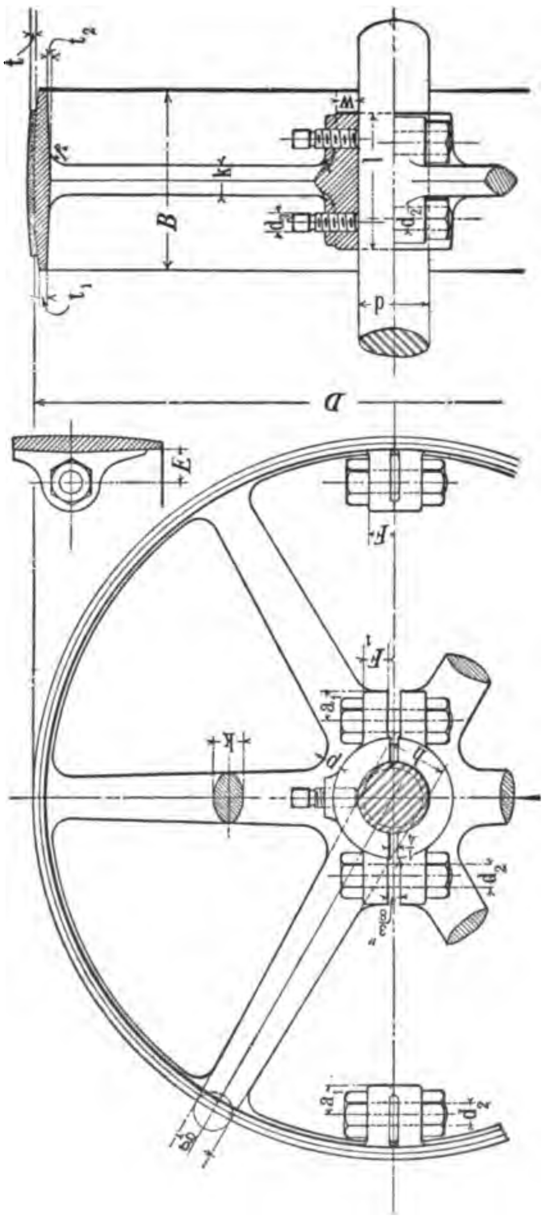


FIG. 174.

d_1 = diameter of set-screw in solid pulley $= \frac{1}{8}d + \frac{7}{16}''$. . (6)

d_2 = diam. of bolt in split pulley at rim and hub = eq. (6)

d_3 = set-screw for key $= .25d$.

D = diameter of pulley.

E = centre of rim bolt from inside of rim $= d_2 + t_2 + \frac{1}{4}''$. (7)

f = radius at end of arms $= \dfrac{K}{2}$

$F = d_2 + \frac{1}{8}''$.

g = width of arm at rim $= \frac{2}{3}h$.

h = width of arm at centre of pulley

$$= \left\{ \begin{array}{l} .6337 \dfrac{\sqrt[3]{BD}}{n} \text{ single belt.} \\ \qquad\qquad \text{(Unwin)} \\ .798 \dfrac{\sqrt[3]{BD}}{n} \text{ double belt.} \end{array} \right\} \quad . \; . \; . \; . \; . \; . \; . \quad (8)$$

k = thickness of arm $= \dfrac{h}{2}$.

l = length of hub $= \frac{2}{3}B$ to B.

p = thickness of rib surrounding hub between arms $= .31\,d$.

t = thickness of belt—see Table 31.

t_1 = thickness of rim $= .6t + .005\,D$. . . . . . . (9)

t_2 = inside taper of pulley rim $= t_1 \div 2$.

$$w = \text{thickness of hub} = \left\{ \begin{array}{ll} .14\sqrt[3]{BD} + \frac{1}{4} \text{ for single belt.} & (10) \\ .18\sqrt[3]{BD} + \frac{1}{4} \text{ `` double ``} & (11) \end{array} \right.$$

R = radius of pulley crown = from 3 to 5 b.

Exercise 84.—A fan revolving with a speed of 1800 rev. per min. develops 8 H.P. and has an 8″ pulley on its shaft. Power is obtained from an engine fly-wheel running at 75 rev. per min. Diam. of fly-wheel = 5 feet. Determine the proper diameters of the intermediate pulleys and make a suit-

able working drawing of the largest of them, similar to Fig. 173 or Fig. 174. *Scale 6″ = 1 foot.*

See Example 2, p. 241.

Wood-split Pulley (Fig. 175).—The Committee on Science and the Arts of the Franklin Institute, in reporting on the Dodge Wood-split Pulley with wooden bushings, stated that in most cases wood-split pulleys are better than iron pulleys. Some of the reasons given for this are as follows:

(1) They are lighter than iron pulleys, lessening the weight on the line shaft and bearings and reducing friction.

(2) The compression fastening of the wooden pulley on iron or steel shafts with wooden bushings will hold the pulley on the shaft quite firmly, dispensing with the use of keys.

(3) The grip of a belt on a wooden pulley exceeds that on an iron pulley to an amount equal to at least 33 per cent.

(4) The method of fastening the wooden pulley to the shaft neither mars nor weakens the shaft, and prevents any tendency to throw the pulley out of balance, as is the case when keys and set-screws are used.

Construction.—"They are built of wooden segments, the face being made of poplar. The two halves of the pulley are secured to the shaft with bolts. The bushings to fit different-sized shafts are made of hard wood, thoroughly air-dried, then bored and kiln-dried; then each bush is counterbored to exact size of shaft, then carefully turned on the outside to fit the bore of the pulley. They are then cut transversely in halves."

Exercise 85.—Make complete working drawings of a wooden split loose pulley 14″ diam., shaft 2″ diam. Projections to be the same as shown in Fig. 175. *Scale 9″ = 1 foot.*

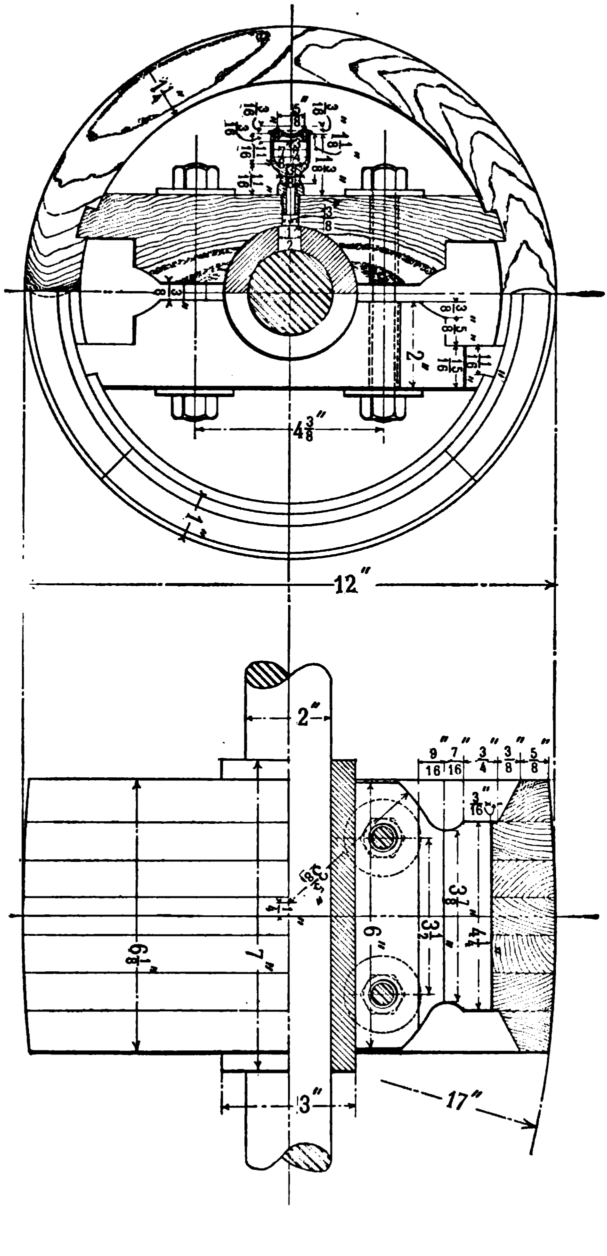


FIG. 175.

All-wrought-steel Pulley.—This pulley as manufactured by the Am. Pulley Co. is shown in Fig. 176. In a paper on the subject by Mr. E. G. Budd before the Franklin Institute in June 1897, the following advantages are claimed for the all-wrought-steel pulley:

(1) They can be used in the heaviest service, clamped to the shaft without keys or set-screws, and never show a sign of slipping.

(2) There is no machining required. The rims and arms

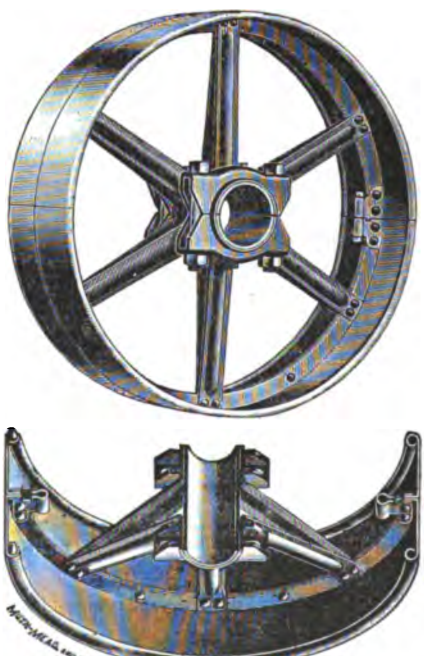


FIG. 176.

are cut with shears and pressed into shape with hydraulic pressure.

(3) Economy of material and symmetry of form, requiring no counterbalance.

(4) Being made of the best and strongest material, it is fully as light as the wood pulley, and much more durable.

Construction.—Referring to Fig. 176 it may be seen that the rim is made up of four segments. It is divided once transversely and once longitudinally. The flanges on the rim at the centre of the face give a means of fastening it to the arms. The rim edges are rolled, giving a neat appearance and preventing the scraping of the belt in throwing it off or on.

The hub is made of half cylinders of heavy steel, and is connected to the rim by a spider divided into four parts, two parts to each half of the pulley. The spider arms are flat and have the edges lying in the direction of rotation. The manner of fastening the arms to the hub and rim, and their corrugated section, as shown at *A* in Fig. 177, make them exceptionally strong for their purpose.

Exercise 86.—Make a true working drawing of the all-wrought-steel pulley shown in Fig. 177 to the dimensions given. *Scale 4″ = 1 foot.*

Cone-pulleys.—In operating machine tools it is often necessary to change power and speed. This is accomplished most easily by means of cone-pulleys. The driven pulley has a series of steps whose diameters are proportioned so that the belt shall fit all pairs of steps with an equal tension, and when the belt is shifted from one pair of steps to another the velocity ratio will be changed.

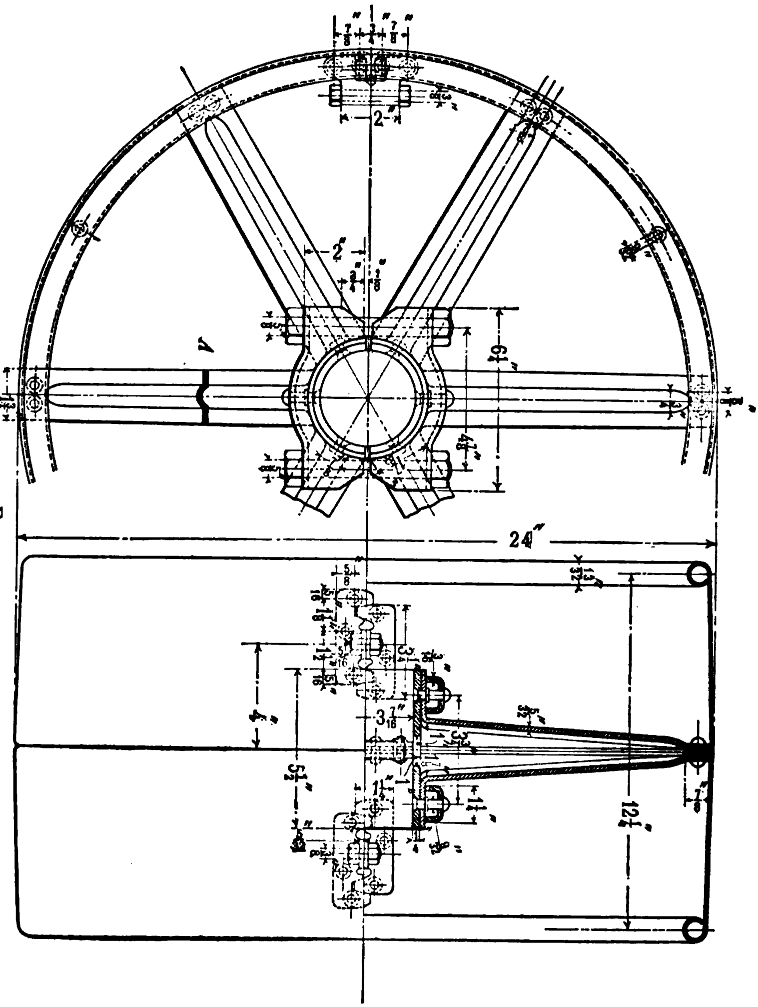


FIG. 177.

Length of Belts (Fig. 178).—

Let L = length of belt;

D = diam. of large pulley;

d = diam. of small pulley;

l = distance between centres of pulleys;

θ = angle whose sine = $\frac{D + d}{2l}$ for crossed belts and

$\frac{D - d}{2l}$ for open belts.

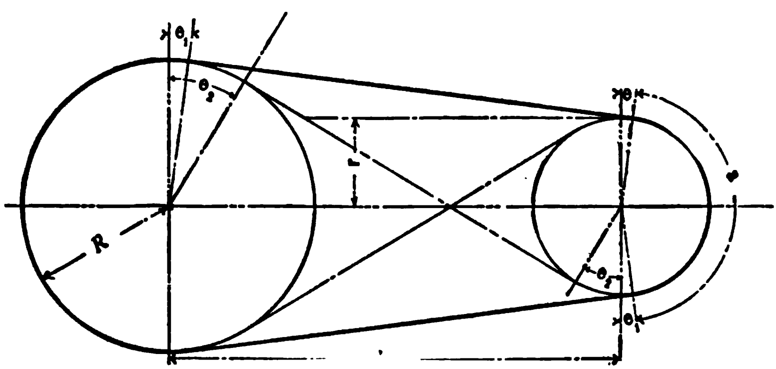


Fig. 178.

From a table of sines find the angle θ in degrees and cos θ.

Then for a crossed belt:

$$L = \left(\frac{\pi}{2} + \theta\right)D + d + 2l \cos \theta \quad . \quad . \quad . \quad . \quad (12)$$

and for an open belt

$$L = \frac{\pi}{2}(D + d) + \theta(D - d) + 2l \cos \theta \quad . \quad . \quad (13)$$

The length of the belt is constant when $D + d$ and l are constant, therefore in designing a pair of cone-pulleys so that the crossed belt will have equal tension on all pairs, it is only necessary to use a pair of equal and similar cones tapering opposite ways.

To design a pair of cone-pulleys for an open belt: Let $D_1D_2D_3D_4$ and $d_1d_2d_3d_4$ = diameters of opposite pulleys (Fig. 179). And using the graphical method given by Mr. C. A. Smith in the A. S. M. E., vol. 10, p. 296, let us suppose the following data to be known:

(1) Diameters of $D_1D_2D_3D_4$ and d_1.

(2) l = distance between centres.

Then let it be required to find the diameters of d_2d_3 and d_4.

C and c are the centres of the opposite cones.

Around centre C draw circles $D_1D_2D_3D_4$ and at centre c draw d_1 to the diameters given.

Draw tangent D_1d_1.

Bisect Cc in the point E and erect a perpendicular EF.

Make the distance $EF = .314l$ found by experiment. With centre F draw arc A tangent to D_1d_1. All lines drawn tangent to arc A will be a common tangent to a pair of cone steps giving the same belt-length as that of the given pair. So to find the diameters of the steps d_2d_3 and d_4 it is only necessary to draw tangents to D_2 and arc A, D_3 and arc A, D_4 and arc A, and with centre c and radii = cd_2, cd_3 and cd_4 respectively, draw the circles of the required steps. This method is an approximation, but close enough for all practical purposes.

Exercise 87.—Referring to Fig. 179: First, assume diameters $D_1 = 18''$, $D_2 = 14''$, $D_3 = 10''$ and D_4 and $d_1 = 6''$, and

find the corresponding diameters of the opposite steps according to Smith's graphical method just explained in connection with Fig. 179.

Second, make complete working drawings of one of the cone-pulleys, showing half longitudinal cross-section and half side elevation combined, and also a half end elevation like Fig. 180. *Scale 6″ = 1 foot.*

PROPORTIONS OF CONE PULLEY.

Let $t =$ thickness of edge of rim $= a$;

$h =$ thickness of hub $= .14\sqrt[4]{BD_1} + \frac{1}{4}''$ from eq. (10);

$H =$ length of hub $= 1.43b$;

$R =$ face radius $= 5B$.

The remaining dimensions may be taken from the following table.

TABLE 82.

(Dimensions in inches.)

b	2	$2\frac{1}{2}$	3	4	5	6	8	10	12	16	18
a	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$
e	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
f	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$
g	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{7}{8}$

Rope Pulleys.—Rope pulleys are made of cast iron with grooved rims, as shown in Figs. 181 and 182. The angle of the groove is usually 45°. The grooves for guide pulleys are semicircular at the bottom, the radius of the curve being a little greater than the radius of the rope. The diameter of a

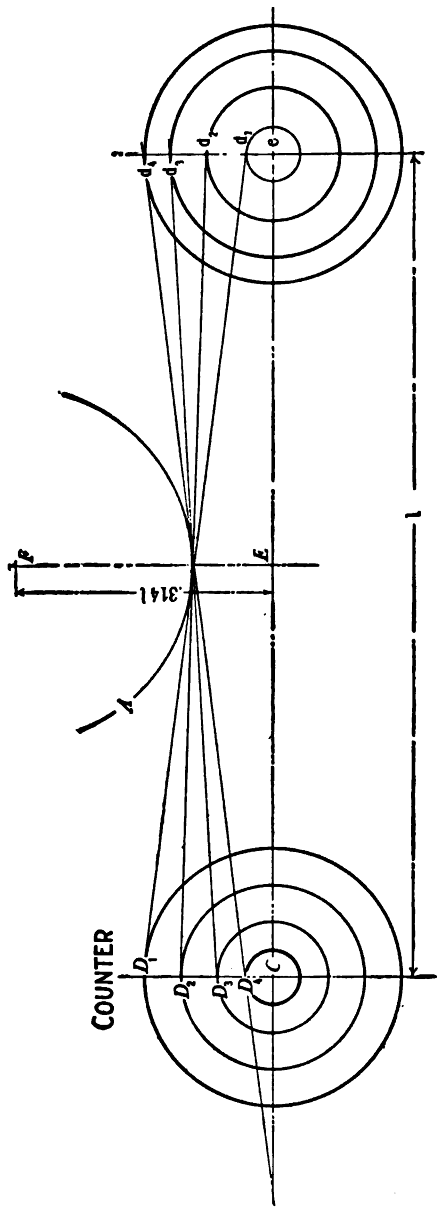


FIG. 179.

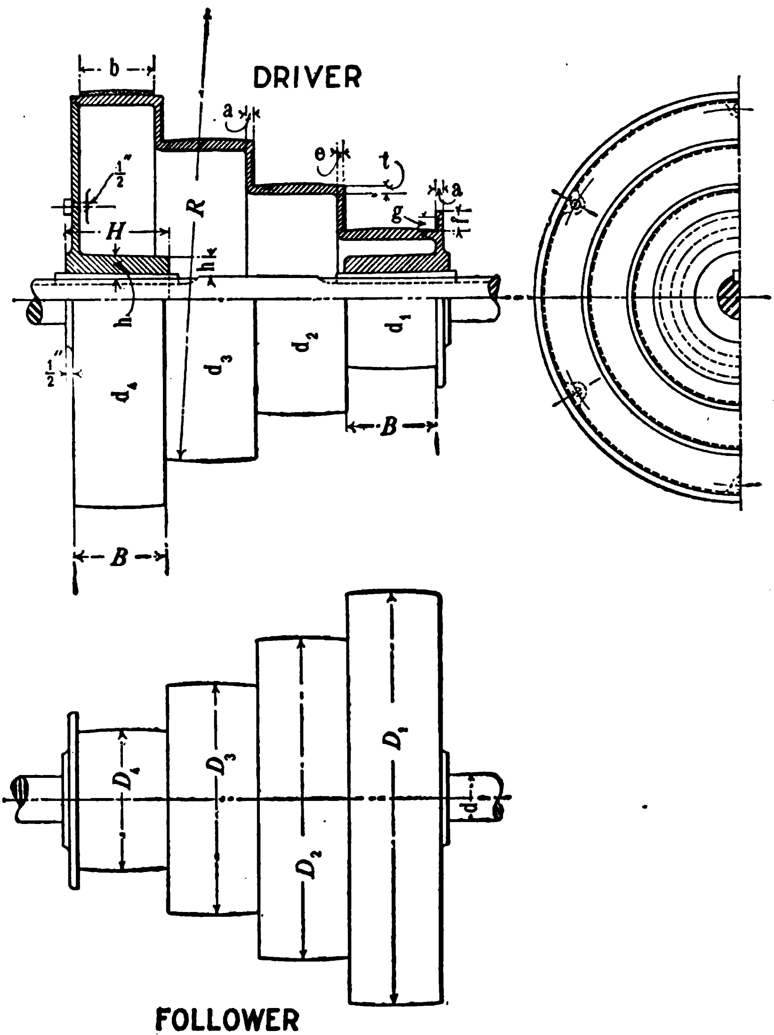


FIG. 180.

rope pulley measured to the centre of the rope should not be less than that given by the following rule:

$$D_1 = (10D + 16)D, \quad \text{where}$$

D_1 = the smallest diameter of the pulley;

D = the diameter of the rope.

As in the case of belt gearing, the slack side of the rope should be on top wherever possible, so as to increase the arc of contact between the rope and the pulley.

Fig. 181. This is the form of groove long used in Great Britain. It has flat sides inclined to each other at from 45° to 60°.

The general practice in America is to use the form of groove shown in Fig. 182, where the sides are curved. This form allows the rope to rotate in the groove, distributing the wear over the entire surface of the rope, making it last longer than it does in the flat-sided groove.

Exercise 88.—Make a drawing of the section of the rim of a rope pulley with five grooves, as shown in Fig. 181. Diam. of rope to be $1\frac{3}{4}''$. *Scale full size.*

Take the other dimensions from the following table.

TABLE 88.

(Dimensions in inches.)

D	*A*	*B*	*C*	*E*	*F*	*G*	*H*
1	7/16	5/16	$1\frac{1}{2}$	1	13/16	3/4	9/16
$1\frac{1}{4}$	1/2	11/32	$1\frac{13}{16}$	$1\frac{1}{4}$	31/32	15/16	11/16
$1\frac{1}{2}$	9/16	3/8	$2\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{8}$	13/16
$1\frac{3}{4}$	5/8	13/32	$2\frac{7}{16}$	$1\frac{3}{4}$	$1\frac{9}{32}$	$1\frac{5}{16}$	15/16
2	11/16	7/16	$2\frac{3}{4}$	2	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{1}{16}$
$2\frac{1}{4}$	3/4	15/32	$3\frac{1}{16}$	$2\frac{1}{4}$	$1\frac{19}{32}$	$1\frac{11}{16}$	$1\frac{3}{16}$
$2\frac{1}{2}$	13/16	1/2	$3\frac{3}{8}$	$2\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{5}{16}$
$2\frac{3}{4}$	7/8	17/32	$3\frac{11}{16}$	$2\frac{3}{4}$	$1\frac{29}{32}$	$2\frac{1}{16}$	$1\frac{7}{16}$

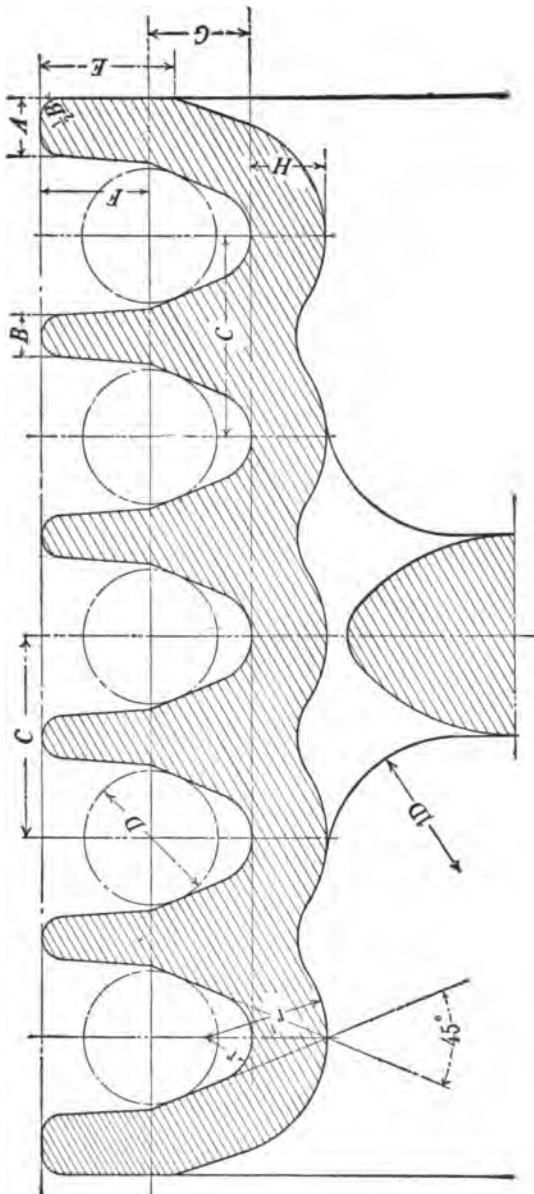


FIG. 181.

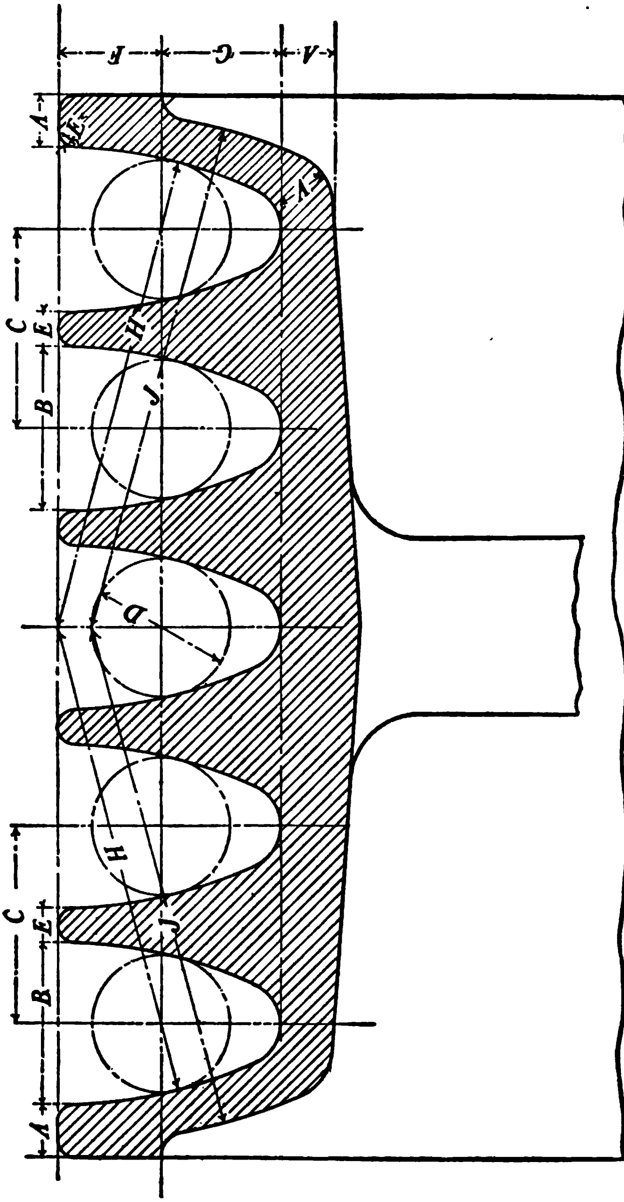


FIG. 182.

Exercise 89.—Make a drawing of the rope pulley rim section shown in Fig. 182. Diam of rope to be $1\frac{1}{2}''$ *Scale full size.* Remaining dimensions may be taken from Table 34.

TABLE 34.

(Dimensions in inches.)

D	*A*	*B*	*C*	*E*	*F*	*G*	*H*	*I*
1	3/8	$1\frac{1}{4}$	$1\frac{3}{8}$	1/4	3/4	7/8	$3\frac{15}{32}$	$3\frac{3}{4}$
$1\frac{1}{4}$	1/2	$1\frac{7}{16}$	$1\frac{3}{4}$	5/16	15/16	$1\frac{1}{8}$	$4\frac{11}{32}$	$4\frac{11}{16}$
$1\frac{1}{2}$	9/16	$1\frac{3}{4}$	$2\frac{1}{8}$	3/8	$1\frac{1}{8}$	$1\frac{5}{16}$	$5\frac{13}{64}$	$5\frac{5}{8}$
$1\frac{3}{4}$	5/8	$2\frac{1}{16}$	$2\frac{1}{2}$	7/16	$1\frac{5}{16}$	$1\frac{1}{2}$	$6\frac{1}{16}$	$6\frac{9}{16}$
2	3/4	$2\frac{3}{8}$	$2\frac{7}{8}$	1/2	$1\frac{1}{2}$	$1\frac{3}{4}$	$6\frac{15}{16}$	$7\frac{1}{2}$
$2\frac{1}{4}$	7/8	$2\frac{11}{16}$	$3\frac{1}{4}$	9/16	$1\frac{11}{16}$	2	$7\frac{51}{64}$	$8\frac{7}{16}$
$2\frac{1}{2}$	15/16	3	$3\frac{5}{8}$	5/8	$1\frac{7}{8}$	$2\frac{3}{16}$	$8\frac{43}{64}$	$9\frac{3}{8}$
$2\frac{3}{4}$	1	$3\frac{5}{16}$	4	11/16	$2\frac{1}{16}$	$2\frac{3}{8}$	$9\frac{17}{32}$	$10\frac{5}{16}$

CHAPTER VIII.

TOOTHED GEARING.

PROPORTIONS OF IRON TEETH. Fig. 183.

p = circular pitch $= k\sqrt{P}$;
p' = diametral pitch $(p \times p') = 3.1416$;
D = pitch diameter $= T \div p'$;
T = number of teeth $= D \times p'$;
l = addendum of tooth $= .3p$;
l' = flank of tooth $= .35p$ to $.4p$;
t = thickness of tooth $= .48p$ for cast-iron teeth,
$= .5p$ for cut teeth;

$k = .04$ for hand-wheels,
$= .05$ for ordinary mill gears,
$= .06$ for wheels of high velocity and mortise gearing;
P = the total force transmitted by one wheel to another through a corner of the tooth $= \dfrac{550H}{V} = 63020\dfrac{H}{RN}$;
V = the velocity of the pitch line in feet per second

$$= \frac{2 \times 3.1416RN}{12 \times 60} = .00873RN;$$

R = the radius of the pitch circle in inches;
N = the number of revolutions of the wheel per minute;
H = the horse-power transmitted by the wheel.

WOOD TEETH or cogs for mortise wheels are usually made thicker than for the iron teeth of the meshing wheel.

t' = thickness of iron teeth to mesh with mortise wheel $= .4p$;

t = thickness of wood cog $= .6p$.

Exercise 90. (Fig. 183.)—*To construct the teeth for a spur gear of 15 teeth and rack, p' or diametral pitch = 2.5. Involute system, angle of action = 15°.*

Draw the centre line C, and compute the diameter of the pitch circle by dividing the number of teeth by p'.

At the point a where the pitch circle cuts C draw line L, making an angle of 15° with the horizontal pitch line, and draw the base circle tangent to L.

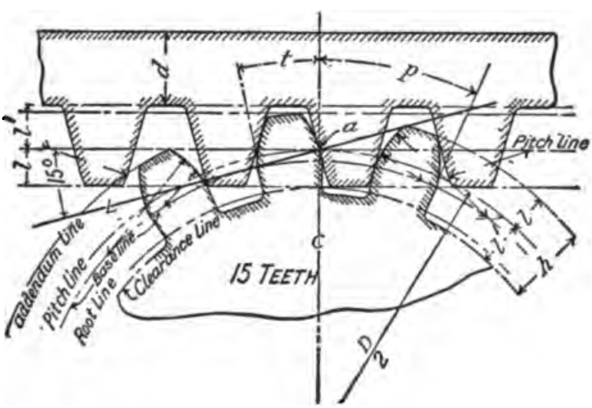


FIG. 183.

To find p: Divide 360° by the number of teeth: the quotient will be the number of degrees in the arc p, which may be laid off with a protractor. Or divide the number of inches in the circumference of the pitch circle by the number of teeth: the quotient will be the pitch. Or divide a quadrant

of the pitch circle with the hair-spring divider into 15 equal parts, and from the point *a* mark every fourth division for the point where the outline of a tooth intersects the pitch circle. Next lay off the thickness of the tooth equal to half the pitch on the pitch circle of the wheel and the pitch line of the rack.

Draw the addendum line of the wheel with a radius $= \frac{D}{2} + .3p.$

The root line of the rack is drawn tangent to the addendum line of the wheel, and the root line of the wheel is tangent to the addendum line of the rack.

To describe the involute curve of the wheel-tooth: Take a piece of tracing-paper or thin celluloid, and trace upon it the straight line *L*, and make a small puncture at the point *a* with a needle. Now at the point where line *L* is tangent to the base line stick a needle, and rotate line *L* about it counter-clockwise until it intersects the base line; at the point of intersection stick another needle, and, removing the first needle, adjust the tracing until the line *L* becomes tangent to the base line at the second needle; then through the puncture *a* in the tracing, with a 4H pencil sharpened to a conical point mark a point on the drawing-paper: this will be a point on the curve. Continue to find similar points until a sufficient number has been found to form the addendum of the tooth.

It will be seen by the figure that the involute curve forming the addendum of the tooth extends below the pitch line to the base line; this part of the curve is generated in a similar way to the part above the pitch line, except that the generating line *L* must be rotated in the opposite direction.

The addendum lines of the other teeth may be traced from the one just found.

The rim of the rack, according to Reuleaux, should not be less than δ in thickness, $= .4p + .125$. Unwin gives $.48p$; Low & Bevis give $.47p$. Use Unwin's proportion.

When the curves have been carefully pencilled as above, they may be inked in with arcs of circles computed by means of the following odontograph table, taken from Geo. B. Grant's "Handbook on the Teeth of Gears":

ODONTOGRAPH TABLE—INVOLUTE TEETH.

CORRECTED FOR INTERFERENCE, INTERCHANGEABLE SET.

Teeth.	Divide by the Diametral Pitch.		Multiply by the Circular Pitch.	
	Face Radius.	Flank Radius.	Face Radius.	Flank Radius.
12	2.70	.83	.86	.27
13	2.87	.93	.91	.30
14	3.00	1.02	.95	.33
15	3.15	1.12	1.00	.36
16	3.29	1.22	1.05	.40
17	3.45	1.31	1.09	.43
18	3.59	1.41	1.14	.46
19	3.71	1.53	1.18	.50
20	3.86	1.62	1.22	.53
21	4.00	1.73	1.27	.57
22	4.14	1.83	1.32	.60
23	4.27	1.94	1.36	.63
25	4.56	2.15	1.45	.70
28	4.82	2.37	1.54	.77
31	5.23	2.69	1.67	.88
34	5.77	3.13	1.84	1.00
38	6.30	3.58	2.01	1.16
44	7.08	4.27	2.26	1.38
52	8.13	5.20	2.59	1.70
64	9.68	6.64	3.09	2.18
83	12.11	8.93	3.87	2.90
115	16.18	12.80	5.16	4.15
200	25.86	22.30	8.26	7.30

For any intermediate number of teeth proportionally intermediate values can easily be found by calculation.

Example.—A gear-wheel has 30 teeth, and the nearest number of teeth in the table is 31; then $\frac{5.23 \times 30}{31} = 5.06$, the number to be divided by p' (1.25), making the true face radius = 4½″ nearly.

The flank of the tooth is radial, and it is joined to the rim with a fillet whose radius is equal to the clearance.

A special rule is provided for the rack-teeth: the flank and one half the face is a straight line drawn at right angles to line L; the other half of the face is a circular-arc centre on the pitch line and a radius found by dividing 2.10″ by p'.

This rounding of the point of the rack-tooth is necessary when it is to mesh with a pinion having less than *28* teeth.

The following tables will be found convenient for comparing the diametral pitch with the circular pitch; they are from Grant's "Teeth of Gears":

Cir. Pitch.	Diam. Pitch.
6	.52
5½	.58
5	.63
4½	.70
4	.78
3½	.90
3	1.05
2¾	1.15
2½	1.25
2¼	1.40
2	1.57
1¾	1.80
1½	2.10
1¼	2.50
1	3.14
¾	4.20
½	6.28

Diam. Pitch.	Cir. Pitch.
½	6.28
¾	4.20
1	3.14
1¼	2.50
1½	2.10
1¾	1.80
2	1.57
2½	1.25
3	1.05
3½	.90
4	.78
5	.63
6	.52
7	.45
8	.39
9	.35
10	.31

Exercise 91. (Fig. 184.)—*To construct the teeth for a spur-gear wheel and pinion;* wheel to have 40 and the pinion 12 teeth. $p' = 2.10$. Walker system, non-interchangeable.

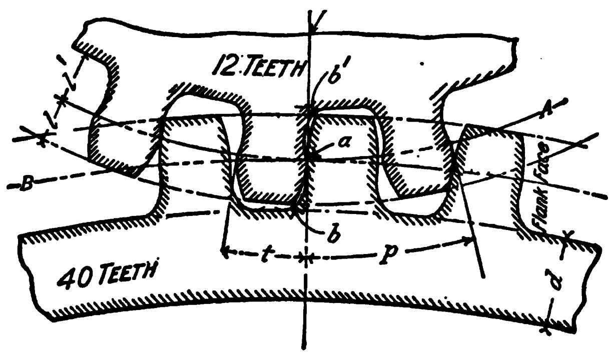


FIG. 184.

The curves of the teeth are epicycloids and epitrochoids, and are found by rolling the pitch circles on each other as follows: For the *addendum* of the wheel-teeth draw arc *A* on a piece of tracing-paper or celluloid, and place it over the drawing tangent to arc *B* at the point *a*. Through the point *a* on the celluloid make a puncture with a needle, and while holding the needle at *a* rotate the celluloid a small distance to the right until arc *A* intersects arc *B*. At the point of intersection place another needle, and, removing the first needle, adjust the celluloid so as to make arc *A* tangent to arc *B* at the second needle, and through the puncture mark a point with the pencil; this will be a point in the curve of the face edge. Other points may be found in a similar way to complete the curves required.

For the face edge of the pinion-tooth roll arc *B* on arc *A*, and the point *a* will describe the curve *ab*.

To draw the *flank* of the wheel-tooth: When arc *A* on the celluloid is tangent to arc *B* at *a*, trace curve *ab* on the celluloid and make a puncture through *b*; then roll arc *A* to

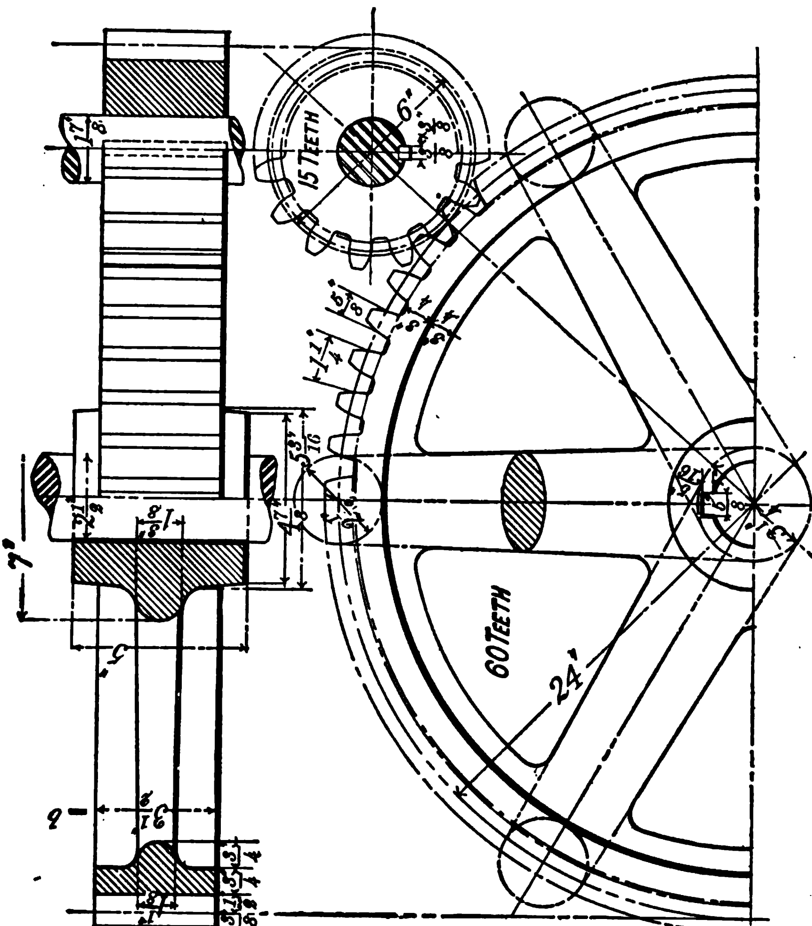


FIG. 185.

the left on *B*, and point *b* will describe the flank of the tooth. The flank of the pinion-tooth is then found by rolling arc *B* on arc *A*, when the point *b′* will describe the curve.

Exercise 92. (Fig. 185.)—*Draw the* HALF ELEVATION, HALF PLAN, *and* HALF SECTIONAL PLAN *of a spur-gear wheel and pinion;* the wheel to have 60 and the pinion 15 teeth. $p' = 2.5$.

Draw all the teeth in one quadrant of the elevation, involute system. Fig 185 is the drawing of a spur-gear wheel made by Messrs. Robert Poole & Sons of Baltimore, Md., and presented to Sibley College for use as a model in the drafting-room.

Exercise 93. (Fig. 186.)—*Draw* ELEVATION, CROSS-SECTION, *and* PLAN *of a bevel-gear wheel and pinion.* The axes are to be at right angles to each other, and the wheel is to have 50 and the pinion 24 teeth. $p' = 2.10$. Radial flank system, non-interchangeable.

Draw centre lines C and C' at right angles to each other, find the radii of the pitch circles, and draw D and D' at the proper distance from the axes. Draw E and E' at right angles to each other. F and F' are the developed pitch circles on which the teeth are drawn, the same as if they were for spur gears. And since the flanks are radial, the rolling circles A and B used to generate the face curves of the teeth are equal in diameter to the radius R and R' of the developed pitch circles of the pinion and wheel respectively.

A model of this wheel will be found in the drafting-room for use in connection with this problem.

Exercise 94. (Fig. 187.) — *Construct a worm-wheel and worm;* the wheel to have 50 teeth, and the worm-teeth to be drawn like those of the involute rack; that is, the face edge will be drawn at right angles to line L, when line L makes

the angle of 15° with the horizontal pitch line *H*, as shown by the longitudinal cross-section in Fig. 187.

The teeth of the wheel are made by a cutter similar to

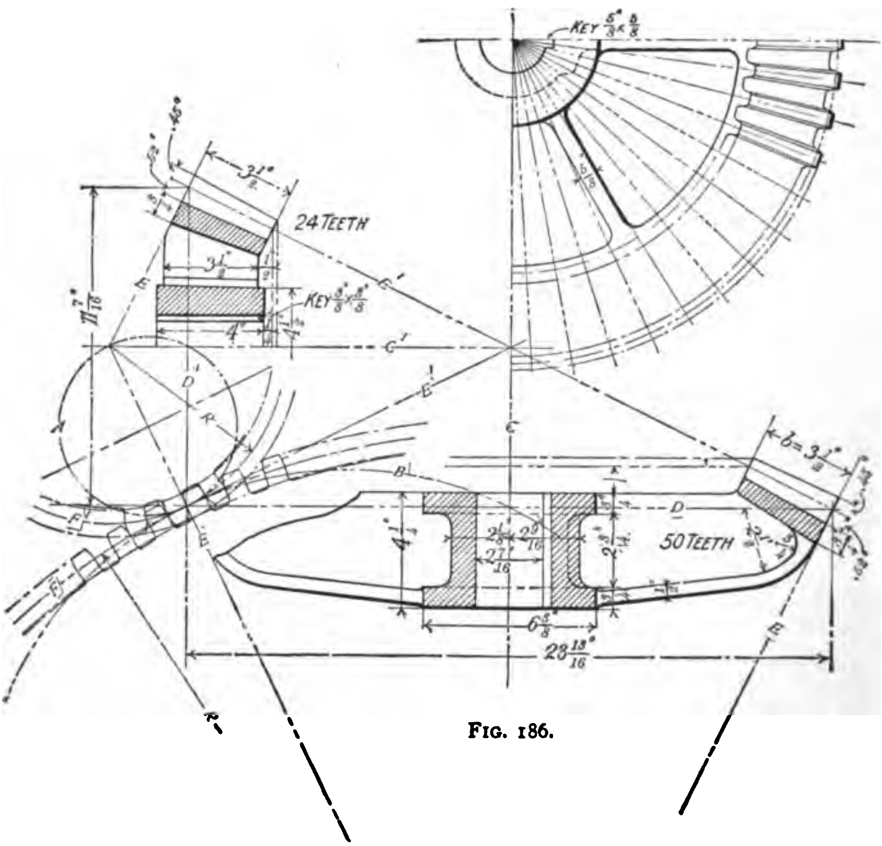


FIG. 186.

the worm, except that grooves are cut in the threads parallel to the axis, and the material is hardened steel. The worm itself is usually made of cast iron, but is sometimes made of wrought iron or malleable cast iron.

The horizontal pitch line should be so placed as to bisect the cross-sectional area of the wheel-tooth at *a*; otherwise the proportions of the teeth may be the same as those used for wheel and rack.

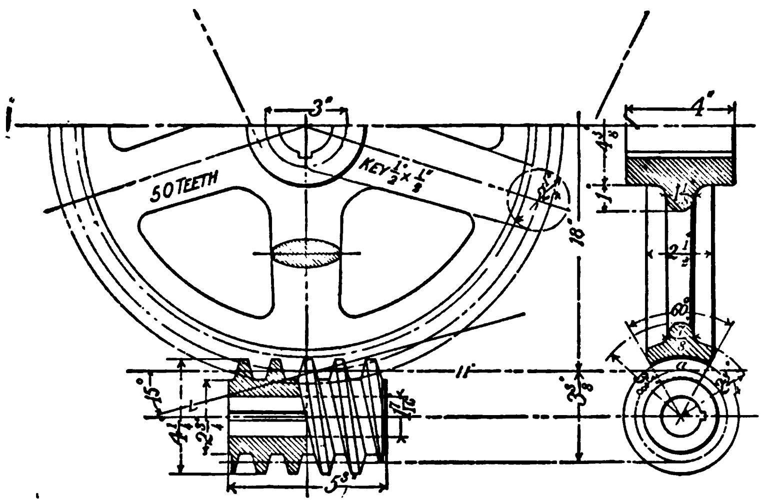


FIG. 187.

Exercise 95. (Fig. 188.) — *Design a cast-iron gear-wheel given the pitch-circle diameter* 51″, *revolutions per minute* 90, *horse-power transmitted* 280.

First find the whole pressure of one wheel on the other $= P = \frac{550H}{V}$; ($V = .00873RN = .00873 \times 25.5 \times 90$;) then find the circular pitch $p = .0447\sqrt{P}$.

The number of teeth can now be found by multiplying the diameter of the pitch circle $D \times 3.1416$, and dividing by $p = \frac{51 \times 3.1416}{p}$ = the nearest *even* number.

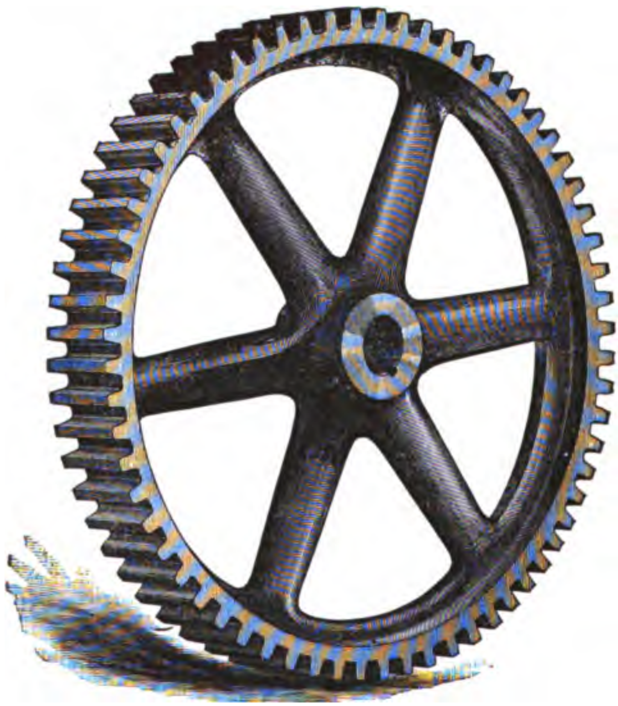


FIG. 188.

Let T represent the number of teeth; then the velocity of the pitch line may be expressed as follows:

$$V = \frac{pTN}{12 \times 16},$$

and the pressure on the teeth is

$$P = \frac{550 \times 12 \times 60 \times H}{pTN} = 396000\frac{H}{pTN}.$$

Taking the width of the teeth into consideration, let

$t = .36p$ for iron teeth when worn,

$= .45p$ for wood teeth when worn;

$h = .7p$ for iron teeth,

$= .6p$ for wood teeth; then

$$P = .046bpf \text{ for iron teeth,}$$
$$= .084bpf \text{ for wood teeth;}$$

and $p = k_1\sqrt{\frac{p}{b}}\sqrt{P}$ when b = width of tooth = from 2 to $4p$,

and in practice

$$k_1 = .0707 \text{ for iron wheels,}$$
$$= .0848 \text{ for mortise wheels.}$$

When $b = 2.5p$, Unwin gives $p = .0447\sqrt{P}$,

$$\text{Low \& Bevis give } p = \sqrt{\frac{P}{200T}}.$$

The dimensions of the teeth may be determined from the proportions already given: b = the breadth of face $= 2.5p$, etc.

As the shaft for this wheel would probably have to resist a combined twisting and bending action, we can assume the diameter of the shaft to be 6″, and the wheel fit 7″.

The *width* and *breadth* of the arms, the *thickness* of the rim, and the *thickness* and *length* of the hub, etc., can be easily determined by the proportions given in the following pages.

Arms of Gear Wheels.—The usual shapes of *arm cross-sections* are shown in Figs. 189 to 192. Fig. 190 is mostly used for pulleys and light wheels; Fig. 191 shows another section that is commonly used in light spur wheels, that in Fig. 192 for heavy spur gears, and that in Fig. 190 for bevel gears.

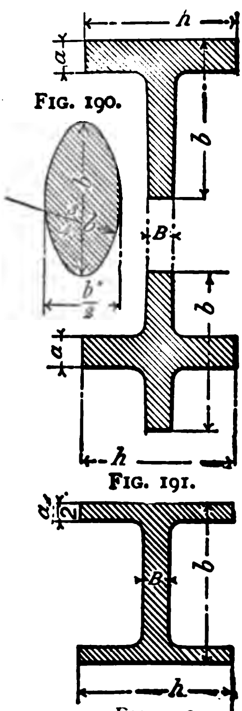


FIG. 189.

FIG. 190.

FIG. 191.

FIG. 192.

When $a = .48p$ = the thickness of the teeth, Unwin gives $h = \frac{.758}{\sqrt{n}}\sqrt{bR}$, measured at the centre of the wheel. Taper $\frac{1}{4}''$ in $12''$ on each side toward the rim. n = the number of arms; R = the radius of the wheel; b = the width of the cross-feathers, which may be = the breadth of the teeth as shown at b in Fig. 193, or $\frac{5}{4}$ the breadth of the teeth measured at the centre of the shaft and from $\frac{3}{4}$ to $\frac{15}{16}$ at the rim.

The ribs or feathers B do not add much to the resistance of the arms to bending in the direction of the driving force, but they are necessary to give lateral stiffness to the arms. Unwin gives $B = .3p$. The feathers should be tapered to facilitate the removal of the pattern from the sand.

To determine the number of arms in a wheel, Low & Bevis give $\frac{D}{36} + 4$. The nearest number divisible by 2 should be tal

Unwin gives four arms for wheels not over 4 ft. in diameter, six arms for wheels of from 4 to 8 ft. in diameter, and eight arms for wheels from 8 to 16 ft. in diameter.

Rims of Gear Wheels.—The usual rim sections are shown in Figs. 193 to 204. The section shown in Fig. 193 is commonly used in light wheels.

The following proportions agree closely with most authorities on the subject: $d =$ the thickness of the rim at the edge $= .48p$. The other proportions are shown in the the figures.

In the rims for bevel gears shown in Figs. 198 to 200 the thickest part of the rim should be $\frac{6}{5}d$.

Figs. 201 and 202 show examples of mortise gears for

FIG. 193. FIG. 194. FIG. 195. FIG. 196.

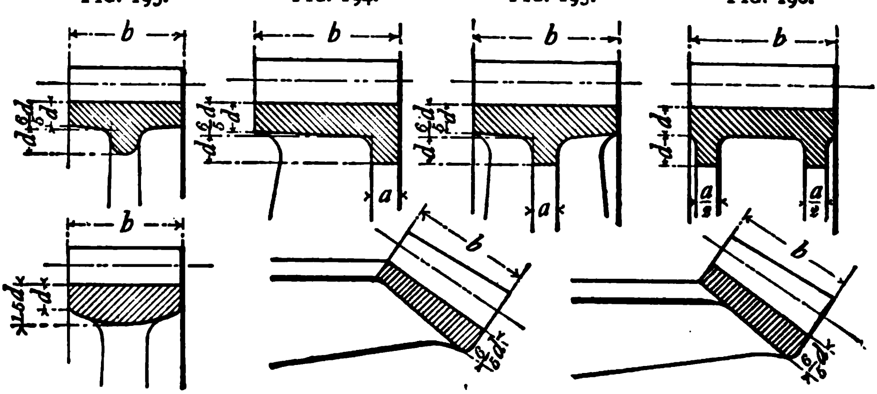


FIG. 197. FIG. 198. FIG. 199.

spur and bevel wheels respectively; the mortise teeth are fixed either by wood keys as shown in Fig. 201, or by round iron pins as shown in Fig. 202. The proportions given in the figures agree closely with good practice.

Shrouding.—When the rim of a wheel is wider than the teeth and extends towards the point so as to form an annular ring uniting the ends of the teeth, the teeth are said to be shrouded. Figs. 203 and 204 give two examples of shrouded teeth. By shrouding out to the pitch circle as shown in Fig. 203, teeth which are no thicker at the root than at the pitch circle can be strengthened about 100 per cent. In the pinion of a pair of gear wheels the shrouding may extend to the points of the teeth as shown in Fig. 204; this compensates for the weak form of the teeth in very small wheels, and prevents their failure from excessive wear.

FIG. 200.

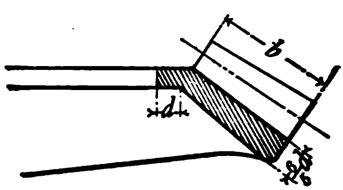
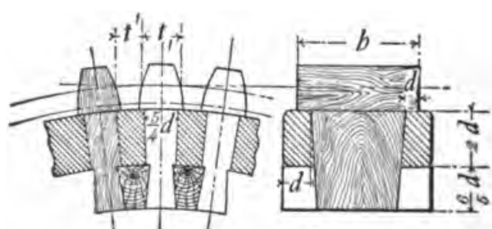


FIG. 201.



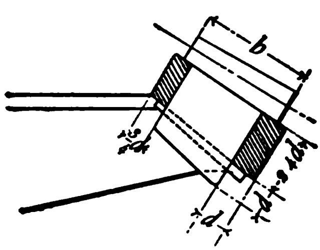


FIG. 202.

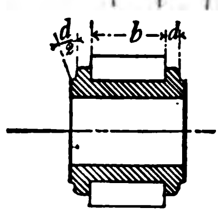


FIG. 203

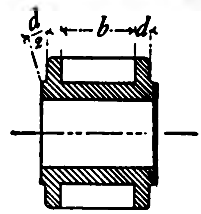


FIG. 204.

Hubs of Gear Wheels.—Figs. 205, 206, and 207 give examples of hubs to correspond to the examples of arms shown in Figs. 189, 191, and 192, respectively.

The thickness of metal surrounding the bore of a gear

wheel is given by Reuleaux $= w = .4h + .4''$ (when $h =$ the width of the arm measured at the centre of the wheel). The keyway should be cut the full length of the hub, and the metal reinforced over the keyway if the wheel is intended for heavy duty. In large wheels the hubs are sometimes strengthened by wrought-iron rings shrunk on both ends; the thickness is made $= \frac{w}{2}$, and the thickness of the metal under the rings is $\frac{3}{4}w$. $b =$ width of teeth.

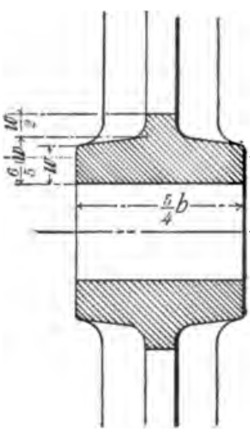


FIG. 205.

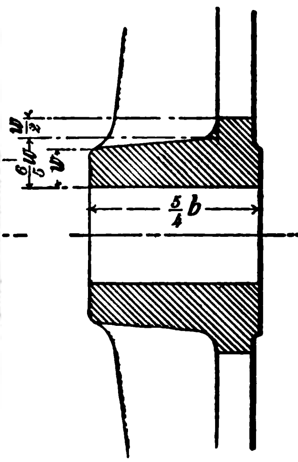


FIG. 206.

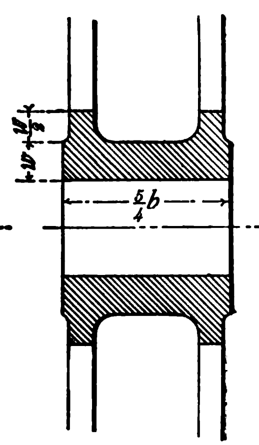


FIG. 207.

In heavy wheels with a large amount of metal surrounding the bore, the hub is sometimes slotted across between the arms to give relief from initial strains due to unequal contraction in cooling; these slots are then filled with metal strips, and the divided hub is held firmly together by the iron or steel ring referred to above.

CHAPTER IX.

VALVES, COCKS, AND OIL-CUPS.

Valves.—A valve is a device for regulating the flow of a fluid through an opening.

Prof. Unwin divides valves into three classes:—(1) Flap-valves, or those which open with a hinge; (2) lift-valves, or those which rise perpendicularly to the seat; (3) slide-valves, or those which move parallel to seat. The valve-face is that part of the valve in contact with its seat when closed.

Foot-valve and Strainer.—Foot-valves are used to hold the water in long suction-pipes; otherwise the pump would have to be charged every time before starting.

The strainer protects the valve from being choked with stones or other solids. The most common foot-valves are made of two cast-iron boxes, called the valve-box and strainer, bolted together by flanges, and having a leather clack-valve between them. The lower box is perforated with circular holes $\frac{1}{4}''$ to $\frac{1}{8}''$ diameter, and is called the strainer or snore-piece. In small foot-valves the suction is generally screwed into the top of the valve-box.

Fig. 207 shows a vertical section and three half plans of a foot-valve for a 9″ suction-pipe. *VB* is the valve-box, *S* the strainer, *A* is the valve-seat, *B* main valve, and *C* an auxiliary

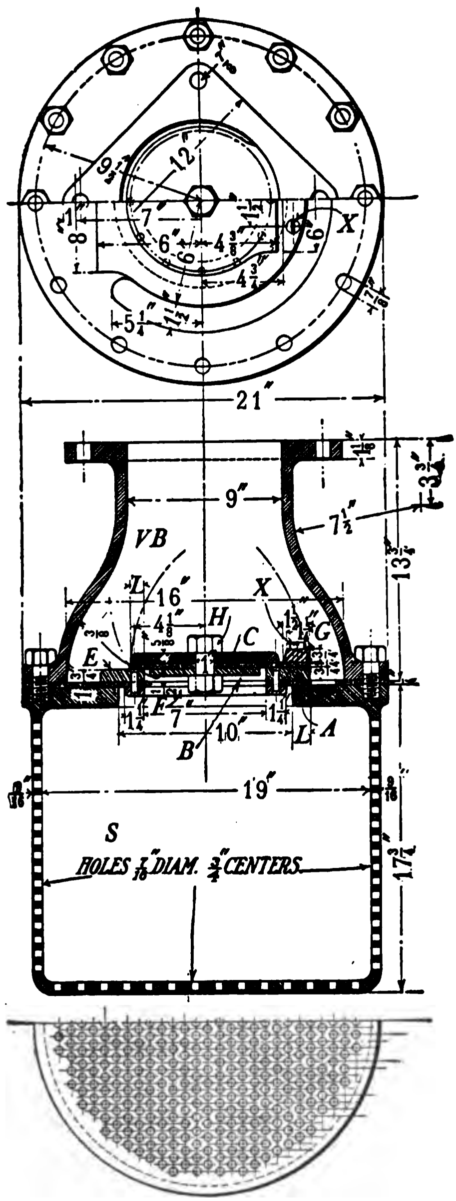

21"
9"
VB
X
16"
H
C
G
E
F
B
10"
L
A
S
HOLES 7/16" DIAM. 3/4" CENTERS
19"
13 3/4"
17 3/4"
7 1/2"

FIG. 207.

valve on top of B. This style of clack is called a *relief* or *break* clack. Mr. Henry Teague, of Lincoln, England, in a paper read before the Inst. of M. E. of England, in 1887, reported having used a 15″ main clack with a 5″ supplementary clack for the purpose of reducing the very great concussion which was had by using the 15″ clack alone, with the result that even when the hand or the ear was placed on the clack-box hardly a tremor or a sound was perceptible. D is the entrance to the suction-pipe.

This double-valve feature gives almost complete freedom from shocks even in large pumps, and therefore works very quietly.

The main valve, made of $\frac{1}{4}''$ leather, forms the joint between the valve-box and the strainer. E is the top and F is the bottom valve-plate, riveted together with $\frac{3}{8}$-inch rivets, and an opening in the centre equal to an area of about one half or one third that of the main opening. This auxiliary opening is fitted with the clack-valve C referred to above. It has an upper and a lower valve-plate, held together with the bolt H and fastened to the main valve with two screws at X, in plan and sectional elevation.

The laps L should be made one tenth of the diameter of the respective valve-openings.

Exercise 96.—Make drawings of foot-valve and strainer shown in Fig. 207, and also an outside elevation of the valve-box and strainer. *Scale 3″ = 1 foot.*

India-rubber Valve.—This valve (Fig. 208) consists of an india-rubber disk D, a brass grating or seat s, and a perforated brass guard. The rubber guard and valve are attached to the grating by a stud-bolt B. The purpose of the guard

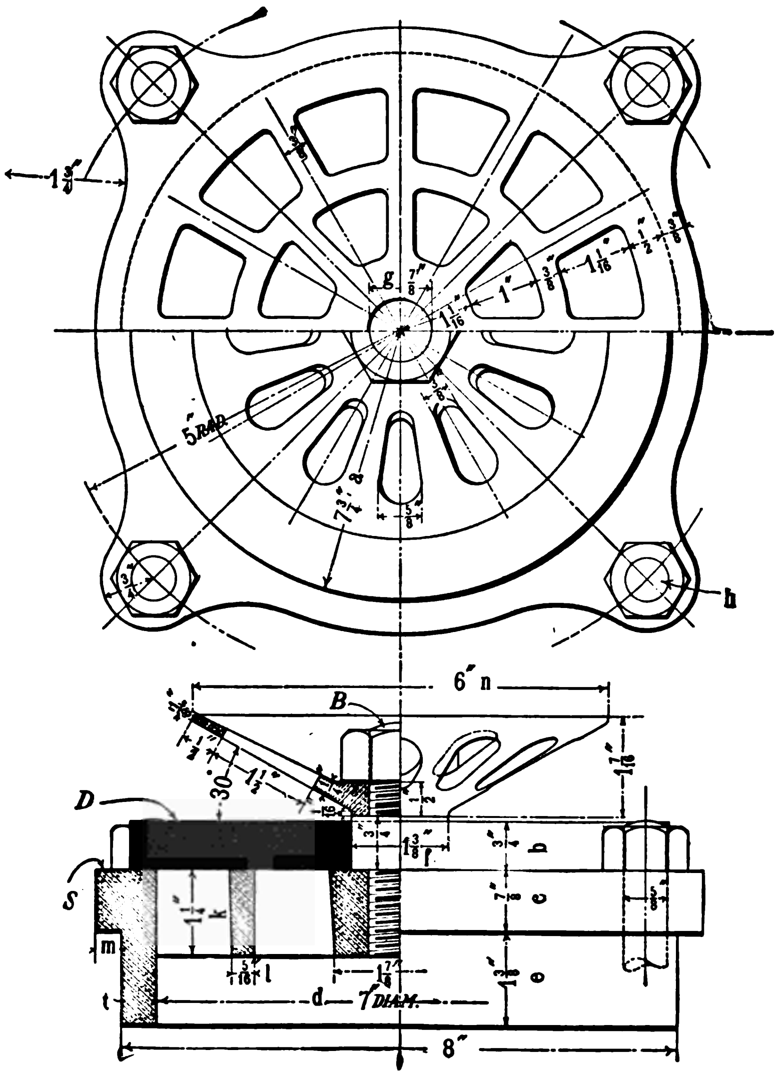


FIG. 208.

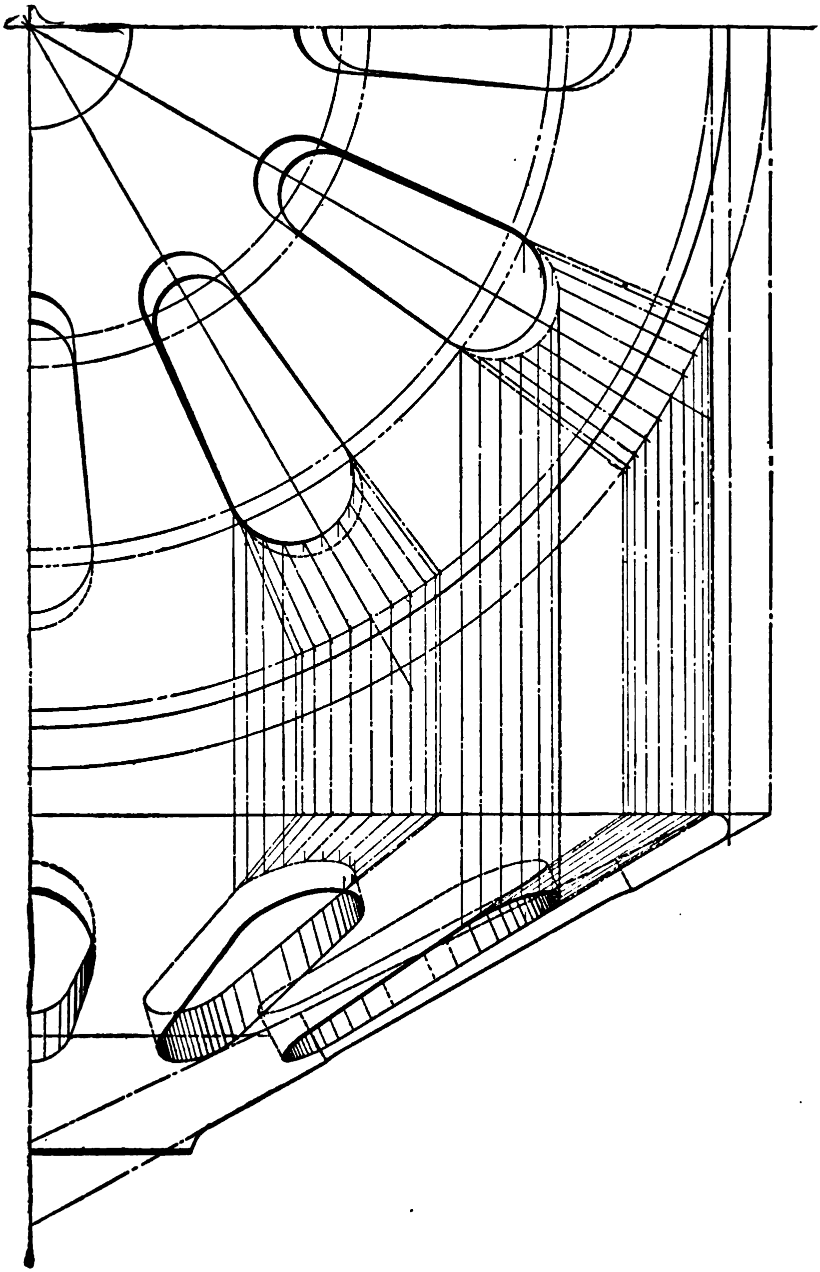


FIG. 209.

is to prevent the valve from rising too high. The perforations in the grating should not be large enough to cause much flexure of the rubber disk. The area of the grating should be such that when the valve is closed the pressure does not exceed 40 lbs. per square inch.

The thickness of the india-rubber disk for large valves—i.e., valves over 6″ in diameter—in condensers and pumps should be $\frac{3}{4}''$ to $\frac{7}{8}''$. India-rubber valves are not good for pressures over 100 lbs. per square inch.

Exercise 97.—Make a complete drawing of the india-rubber valve as shown in Fig. 208. *Scale full size.* The projection of the perforations in the conical guard is shown in Fig. 209.

The following proportions represent good practice. Use the nearest $\frac{1}{16}''$. Unit $= .19\sqrt{d}$.

a = diameter of india rubber disk	=	15.5	of unit.
b = thickness of the india-rubber disk	=	1.6	"
c = thickness of the grating-lip	=	1.75	"
d = diameter of the valve.			
e = depth of seat-body	=	2.75	"
f = diameter of stud-body	=	2.75	"
g = diameter of stud	=	1.75	"
h = diameter of holding-down bolt	=	1.25	"
k = depth of grating	=	2.50	"
l = thickness of grating-rib	=	.65	"
m = width of seat-lip	=	.75	"
n = diameter of guard	=	12.00	"

Exercise 98.—Make a complete drawing of an india-rubber disk-valve similar to Fig. 208. $d = 10''$. *Scale 9″ = 1 foot.*

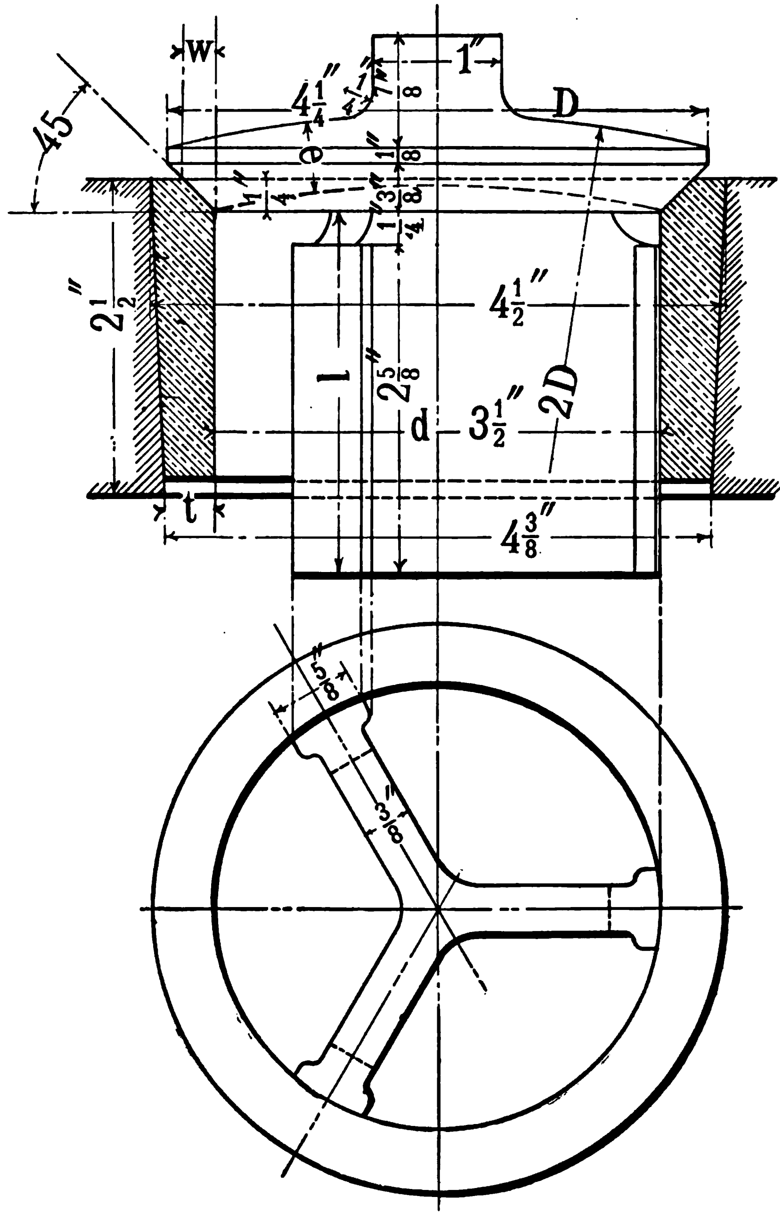


FIG. 210.

Lift- or Wing-valves (Fig. 210).—These valves are usually made of brass. The essential features are a circular disk and seat. The edges between the disk and seat are bevelled to the angle of 45°, and are easily fitted and ground together. Springs or rods are used to close these valves when it is necessary to place them in a horizontal position. To give the valve a partial rotation and provide a new seating at each stroke the wings are curved slightly, as shown at Fig. 211. The curving is arbitrary, and may be projected as shown in the figure. The outside of the seat has usually a taper of $\frac{1}{2}''$ in 12″, but is sometimes driven straight. The amount of the lift of the valve may be determined as follows:

Let

a = area of opening in seat;
d = diameter of opening in seat;
L = lift of valve.

Then

$$a = .7854d^2 \text{ and } L = .35d. \quad \ldots \ldots \quad (1)$$

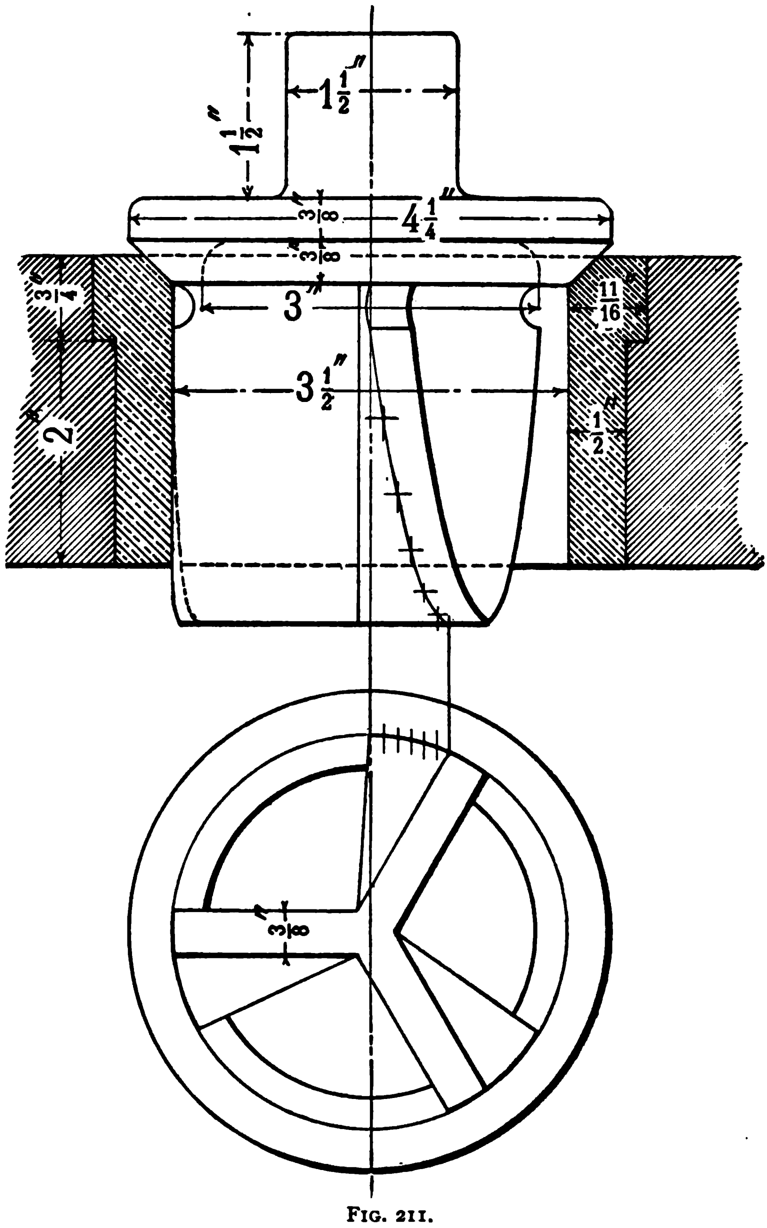
Taking a unit of proportion $= .2\sqrt{d}$, then

e = thickness of disk = 1.3;
l = length of wings = 8;
t = thickness of seat = 1 at small end.

Exercise 99.—Draw the valve as shown in Fig. 210 to the dimensions given. *Scale full size.*

Exercise 100.—Make drawing of the curved wing-valve as shown in Fig. 211. *Scale full size.*

Spindle-valves (Fig. 212).—These valves are guided centrally by means of a spindle and bridge; otherwise they are



$1\frac{1}{2}''$
$1\frac{1}{2}''$
$\frac{3}{8}''$
$4\frac{1}{4}''$
$\frac{3}{8}''$
$3''$
$\frac{3}{4}''$
$\frac{11}{16}''$
$3\frac{1}{2}''$
$2''$
$\frac{1}{2}''$
$\frac{3}{8}''$

FIG. 211.

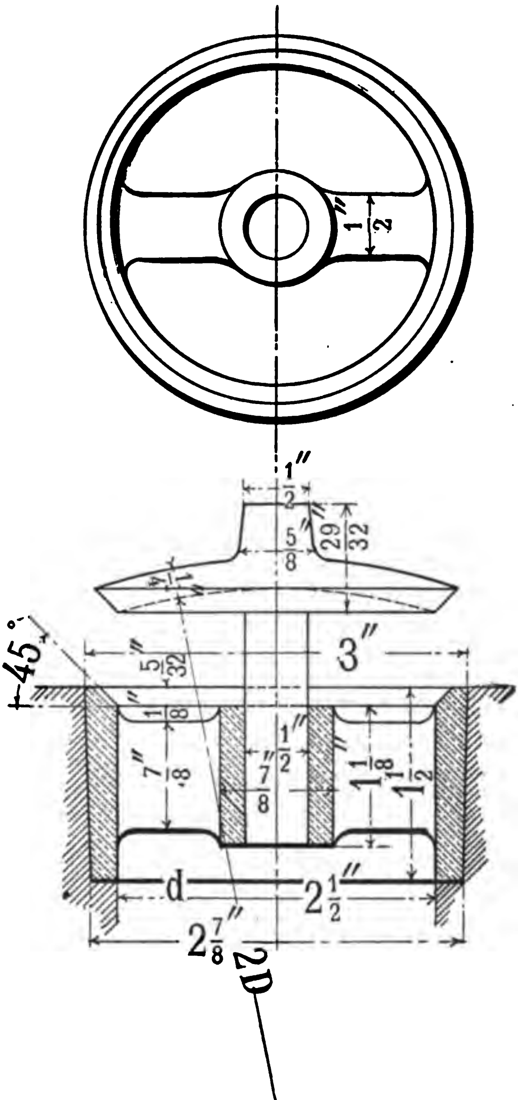


FIG. 212.

similar to the wing-valve, but used for light work in pumps. The wing-valve and the spindle-valve are sometimes made with a flat seat and a leather face and also used for light duty in pumps, but have no advantage over the bevelled metal edges.

Let W (Fig. 210) = the width of the bearing-edges measured perpendicularly to the axis of the valve, p = the maximum difference of pressure on the two sides of the valve; then $\frac{p}{\pi d W}$ = the crushing pressure per square inch on the narrow bevelled edges of the valve and seat.

The greatest safe pressure per square inch for phosphor-bronze is 3000 lbs.; for gun-metal, 2000 lbs.; cast iron, 1000 lbs.; and leather and india-rubber, 700 lbs.

Exercise 101.—Make drawings of the spindle and valve as shown in Fig. 212. *Scale full size.*

Ball-valves (Fig. 213).—These valves are much used in deep well-pumps and small fast-running pumps. To guide the lift of the ball it is surrounded by a cage with three or four ribs. The ribs should be as narrow as safety will permit, so as not to interfere with the free flow of the fluid above the valve-seat. Gun-metal is the best material for the balls. To lighten them they should be made hollow.

The usual proportions for the ball-valves are given below:

$$\text{Unit} = .2\sqrt{d}.$$

a = diameter of ball	$= 1.34d$.
b = inside diameter of seat-casing	$= 1.12d$.
c = thickness of ball-guide	= .9 times unit.
e = distance between guides	$= a + \frac{1}{16}''$.

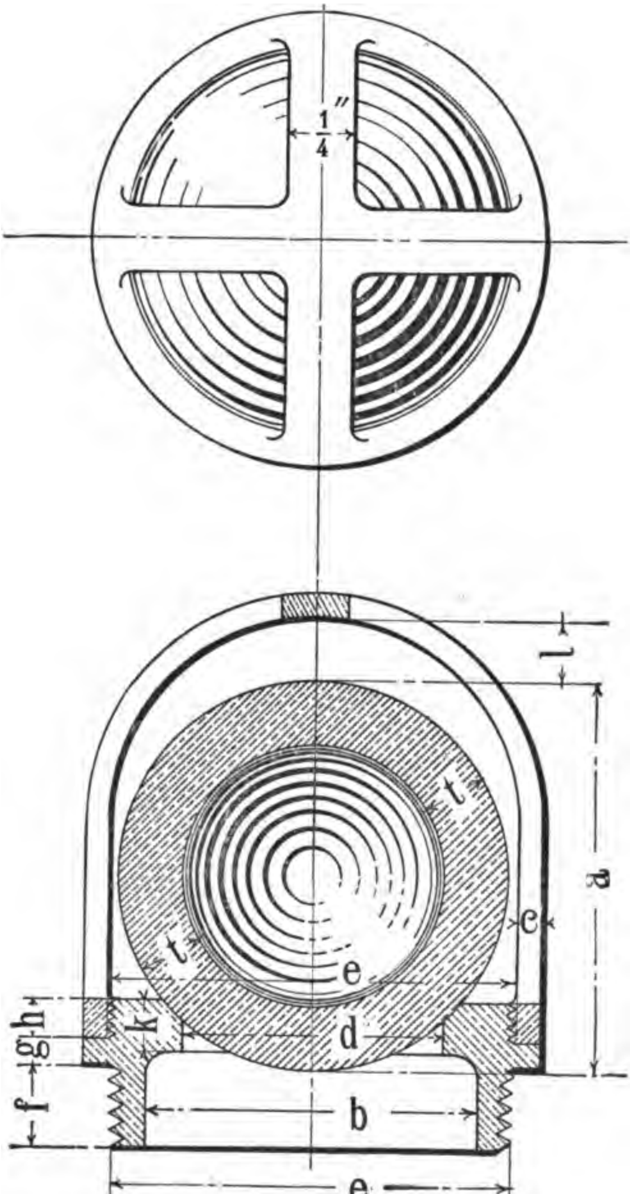


Fig. 213.

f = length of seat-shank	= 3 times unit.	
g = thickness of seat-flange	= 1	"
h	= 1.2	"
k	= 1.8	"
l = lift of valve	= .16d	"
t = thickness of ball-shell	= 1.2	"

These valves work best with a small lift. William M. Barr says that the lift of ball-valves should not exceed ¼″.

Exercise 102.—Make drawings similar to those shown in Fig. 213. $d = 1\frac{1}{2}''$. *Scale 1½ full size.*

Flat India-rubber Disk-valves.—Fig. 214 shows an ordinary example of this style of valve for cold water. The valve-seat and spindle are cast in one piece. The spindle is turned and polished, and the hole in the india-rubber disk is $\frac{1}{16}''$ larger than the diameter of the spindle. This allows free action of the valve. The valve-seat is screwed into place with a pitch of eight threads to the inch, which may be maintained for all sizes up to 4½″ diameter. Mr. W. M. Barr gives the following dimensions for india-rubber valves:

TABLE 85.

Diameter.	Thickness.	Hole.
2″	⅜″	½″
2½″	$\frac{7}{16}''$	½″
3″	½″	$\frac{9}{16}''$
3½″	⅝″	⅝″
4″	⅝″	1⅛″
4½″	¾″	1⅜″
5″	¾″	$1\frac{3}{16}''$

Springs give good results if made with No. 12 brass wire for 2″ and 2½″ valves; No. 10 wire for 3″ and 3½″ valves,

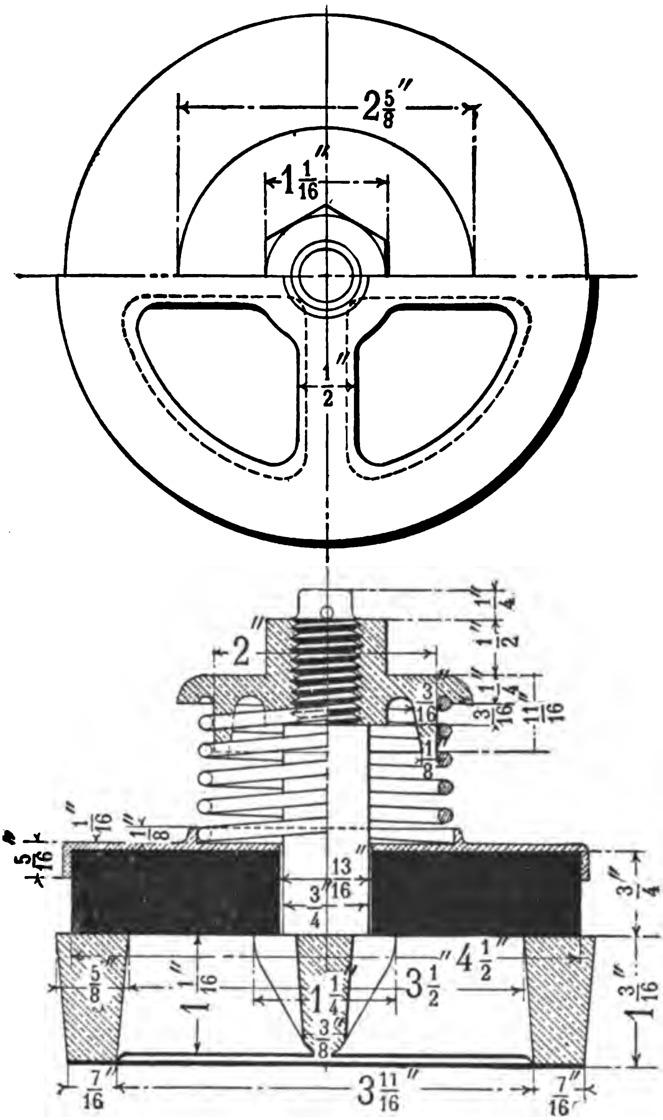

$2\frac{5}{8}''$
$1\frac{1}{16}''$
$\frac{1}{2}''$
$2''$
$\frac{1}{4}''$
$\frac{1}{2}''$
$\frac{1}{4}''$
$\frac{3}{16}$
$\frac{3}{16}$
$\frac{11}{16}$
$\frac{1}{8}$
$\frac{1}{16}''$
$\frac{1}{8}''$
$\frac{5}{16}$
$\frac{13}{16}''$
$\frac{3}{4}''$
$\frac{3}{4}''$
$4\frac{1}{2}''$
$\frac{5}{8}$
$1\frac{1}{16}''$
$1\frac{1}{4}''$
$3\frac{1}{2}$
$\frac{3}{8}''$
$1\frac{3}{16}''$
$\frac{7}{16}''$
$3\frac{11}{16}''$
$\frac{7}{16}''$

FIG. 214.

and No. 8 for 4″ and $4\frac{1}{2}$ valves. The outside diameter of the spring may be = .5 that of the valve-disk. Five to six coils will give a suitable elasticity.

Exercise 103.—Make drawings for the india-rubber flat-disk valve, as shown in Fig. 214, to the dimensions given. *Scale full size.*

Globe-valves.—These valves are opened and closed by hand. The valve in Fig. 214 is for steam. When such a valve is used for cold water the valve-face is made of leather or india-rubber, and when for hot water the india-rubber is mixed with graphite.

The construction of the valve is so plainly shown in Fig. 215 that a description seems unnecessary.

Exercise 104.—Make drawings of globe-valve as shown in Fig. 215, and also a right-end elevation. *Scale full size.*

Stop-valve (Fig. 216).—This is another style of lift-valve controlled by hand. The particular valve shown in the figure is used as a throttle-valve by the Ball Engine Co., who kindly sent drawings.

Let

t = thickness of casing;
p = pressure in lbs. per square inch;
d = diameter of the sphere in inches;
f = safe bursting strength of material.

Take 2000 for cast iron and 17,500 for yellow brass, and use a factor of safety 8, which gives 2500 for the former and about 2200 for the latter; then

$$t = \frac{pd}{4f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

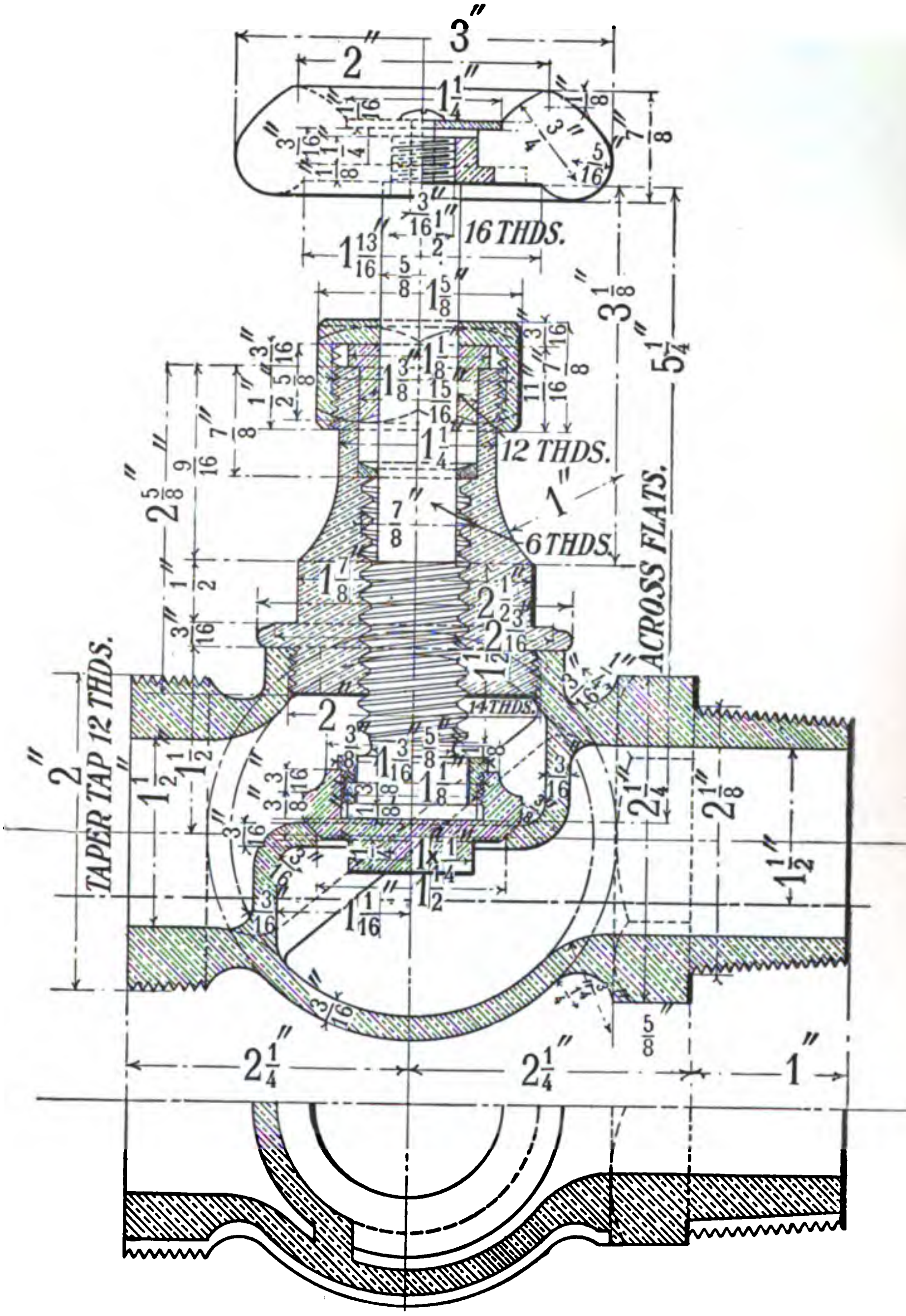


FIG. 215.

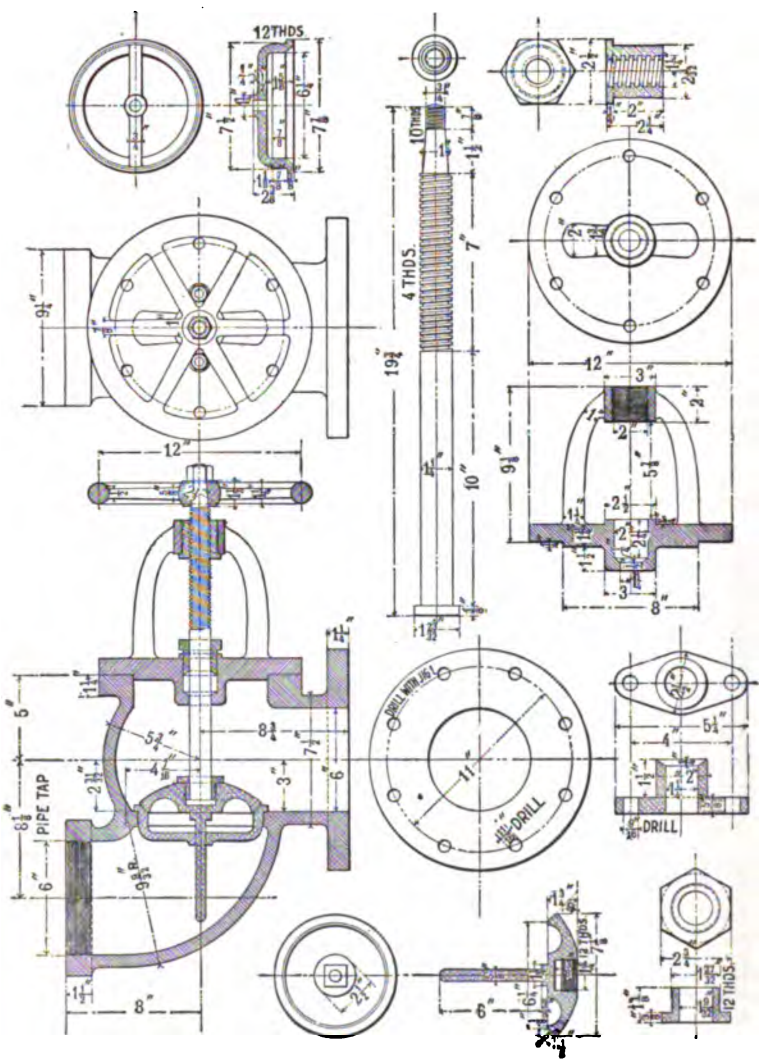


FIG. 216.

The lift of the valve may be determined by formula (1) for winged lift-valves. The valve and its seat must pass through the valve-chest, so the opening should be made about $\frac{1}{8}''$ larger than the outside diameter of the valve-seat.

The length of the thread on the valve-stem is equal to the length of the nut + lift of valve + $\frac{1}{4}''$ for clearance.

Exercise 105.—Make drawings of the stop-valve as shown in Fig. 216. *Scale 4″ = 1 foot.*

Make the diameter of the inlet $6\frac{3}{4}''$ to the root of the thread, instead of 6″ as shown in the figure.

Boiler Check-valve.—Fig. 217 shows working drawings of the Foster Safety Boiler-check.

Exercise 106.—Make drawing of the Foster Boiler-check as shown in Fig. 217. Scale, full size.

Cocks.—Cocks are valves which operate with a rotary motion. The most common style of cock is that which consists of a plug made in the form of a truncated cone rotating in a seat of the same shape cast on a pipe.

In Fig. 218 *P* is the plug, and *C* the casing or conical seat. *O* is the opening through the plug. By rotating the plug in one direction the openings are brought in line with the inlet *A* and outlet *B* of the pipe or casing. In this position the cock is open. Further rotation through 90° in either direction will bring the openings in the plug opposite the solid parts of the casing and close the valve.

Exercise 107.—Make drawings of the *blow-off* cock shown in Fig. 218, and in addition to the views given make a ***half sectional plan*** and ***half sectional end view.*** Scale, full size.

In Fig. 219 is shown a blow-off cock which is really a *wing-valve*, opened and closed by a piston which in turn is op-

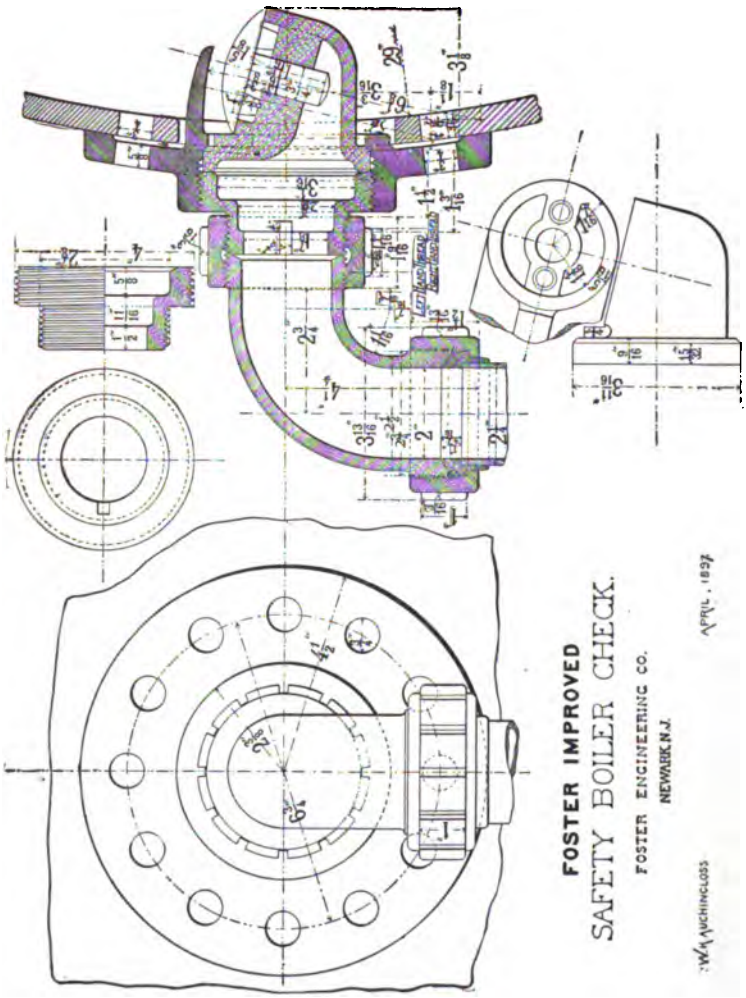

FOSTER IMPROVED
SAFETY BOILER CHECK.
FOSTER ENGINEERING CO.
NEWARK, N.J.
APRIL, 1897
W.K. AUCHINCLOSS

FIG. 217.

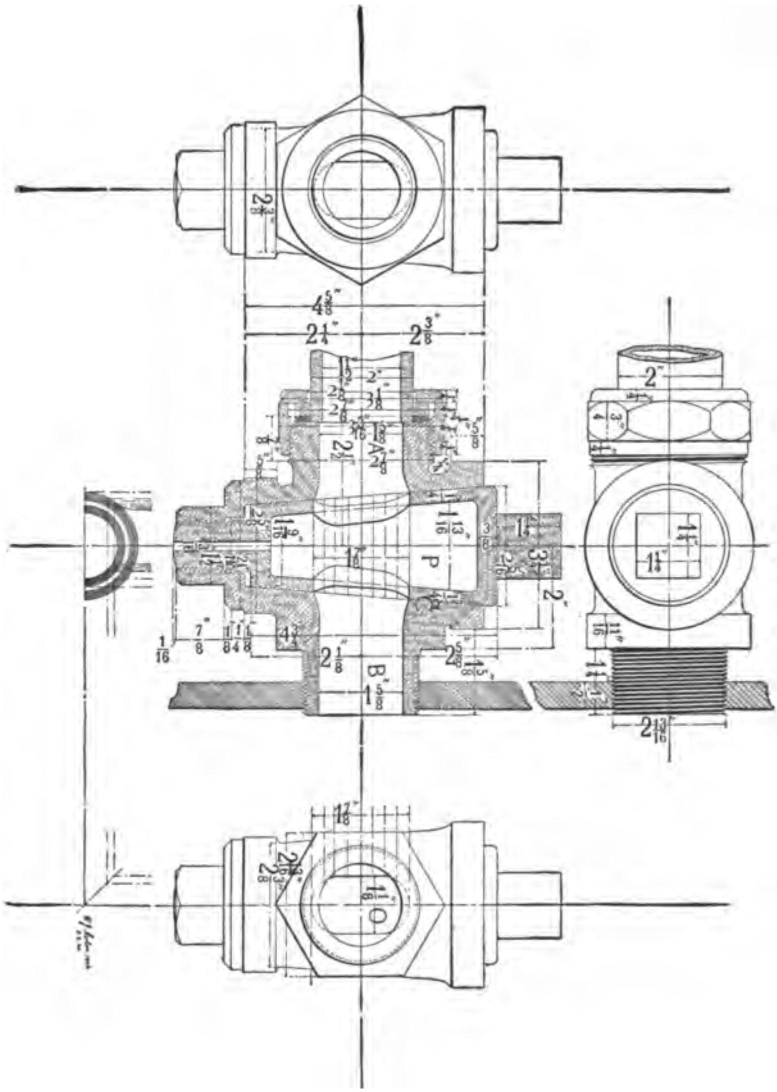


FIG. 218.

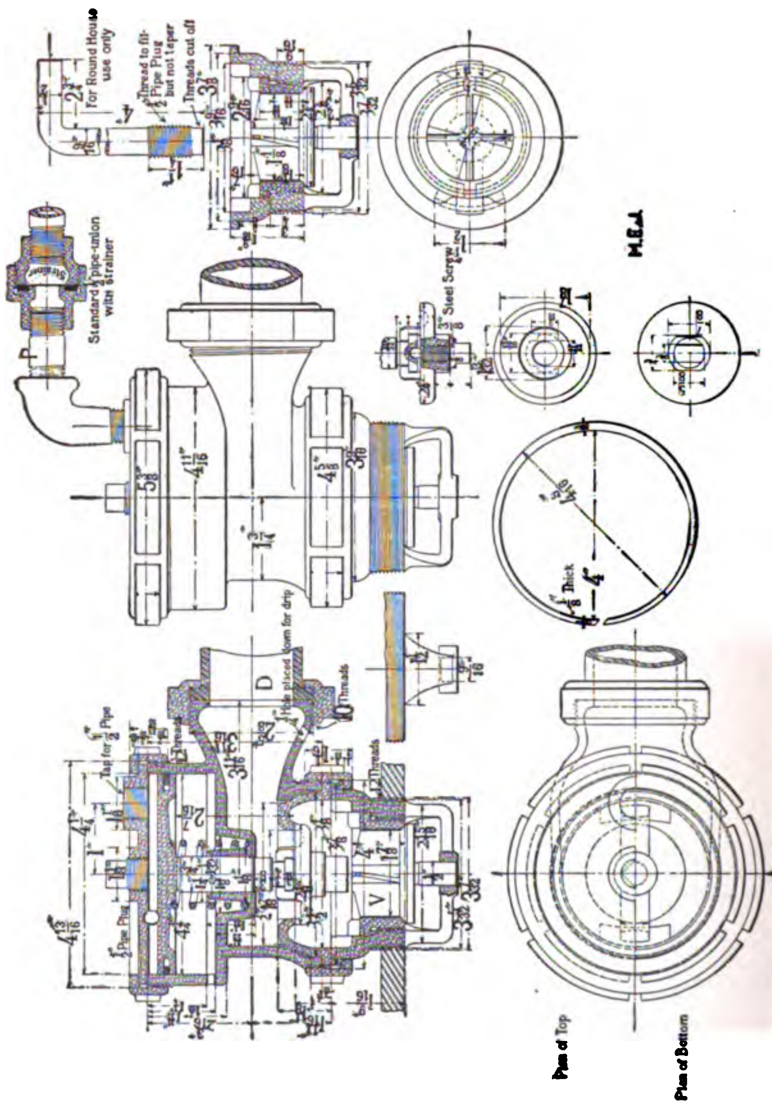

Standard 1/2" pipe-union with strainer
For Round House use only
Thread to fit 1/2" Pipe Plug but not taper
Threads cut off
Steel Screw
Tap for 1/2" Pipe
1/2" Pipe Plug
Hole placed down for drip
10 Threads
12 Threads
Thick
Plan of Top
Plan of Bottom

Fig. 219.

erated by means of compressed air. The wing-valve V is held on its seat by the steam-pressure in the boiler. When compressed air is introduced into the cylinder C through the pipe P the piston is pushed against the valve, opening it and allowing the contents of the boiler to blow through the cock into the discharge-pipe D.

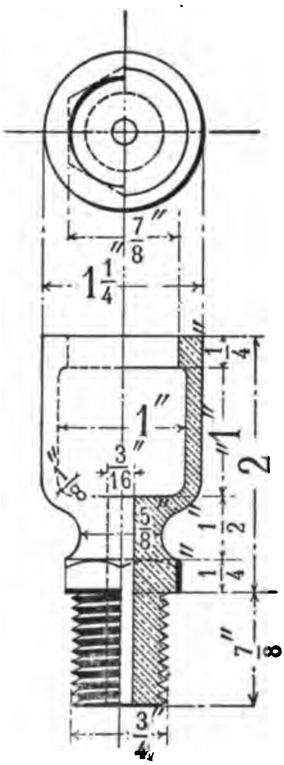


FIG. 220.

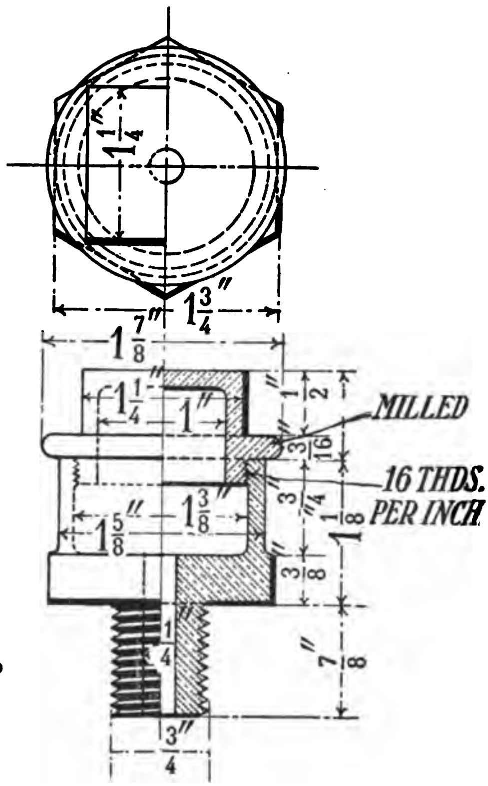


FIG. 221.

Exercise 108.—Make complete drawings as shown in Fig. 219. Scale, full size.

Oil-cups.—There are many forms of oil-cups. Figs. 220

to 225 inclusive show the construction of some of the oil-cups used in the locomotives of the Lehigh Valley Railway.

Fig. 220 is one of the simplest forms of oil-cups. The material is brass, cast in one piece. When charged, the reser-

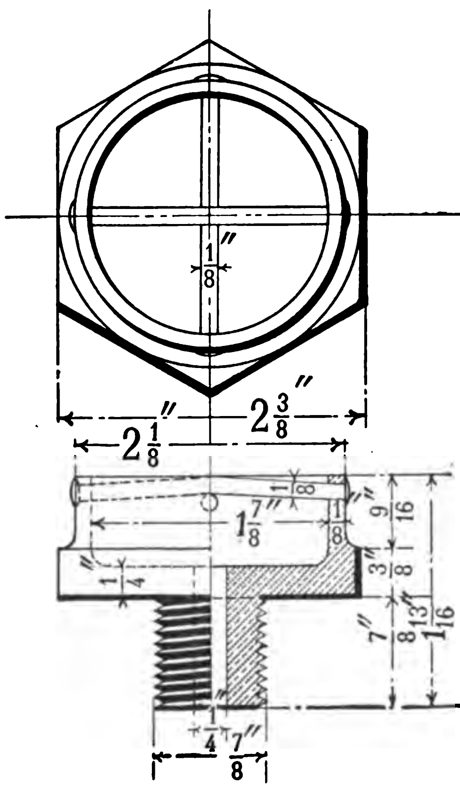


FIG. 222.

voir is filled with waste and oil. This cup is used on the link-hanger.

Fig. 221 shows another simple form of oil-cup, used to oil the rocker-box and cross-head.

Fig. 222 is a drawing of the oil-cup for the main rod, front end; cross-wires prevent the waste from being thrown out.

Fig. 223 shows another form of oil-cup used on the valve stem. The flow of the oil is regulated by the spindle *S*, and

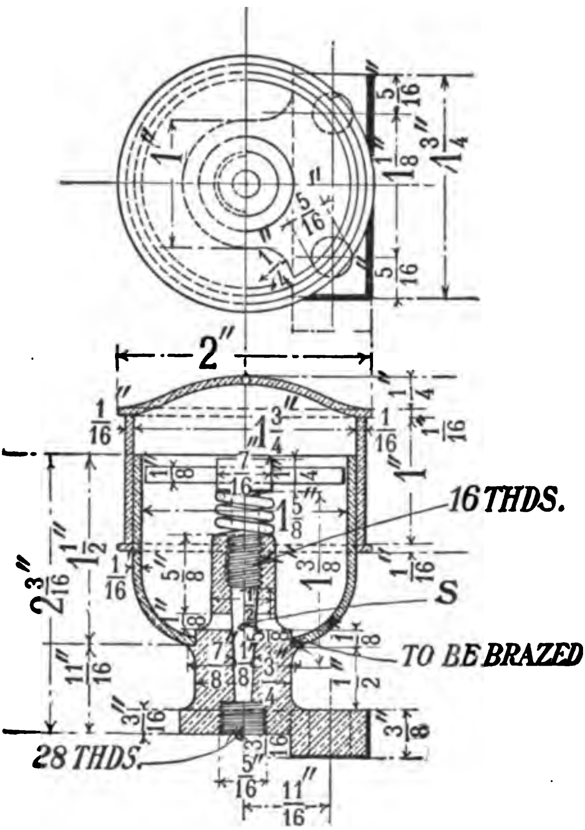


FIG. 223.

the duty of the spring is to hold it in position. This is made of $\frac{1}{16}''$ brass wire $\frac{1}{2}''$ long when unloaded.

Fig. 224 gives a form of oil-cup for the front end of the

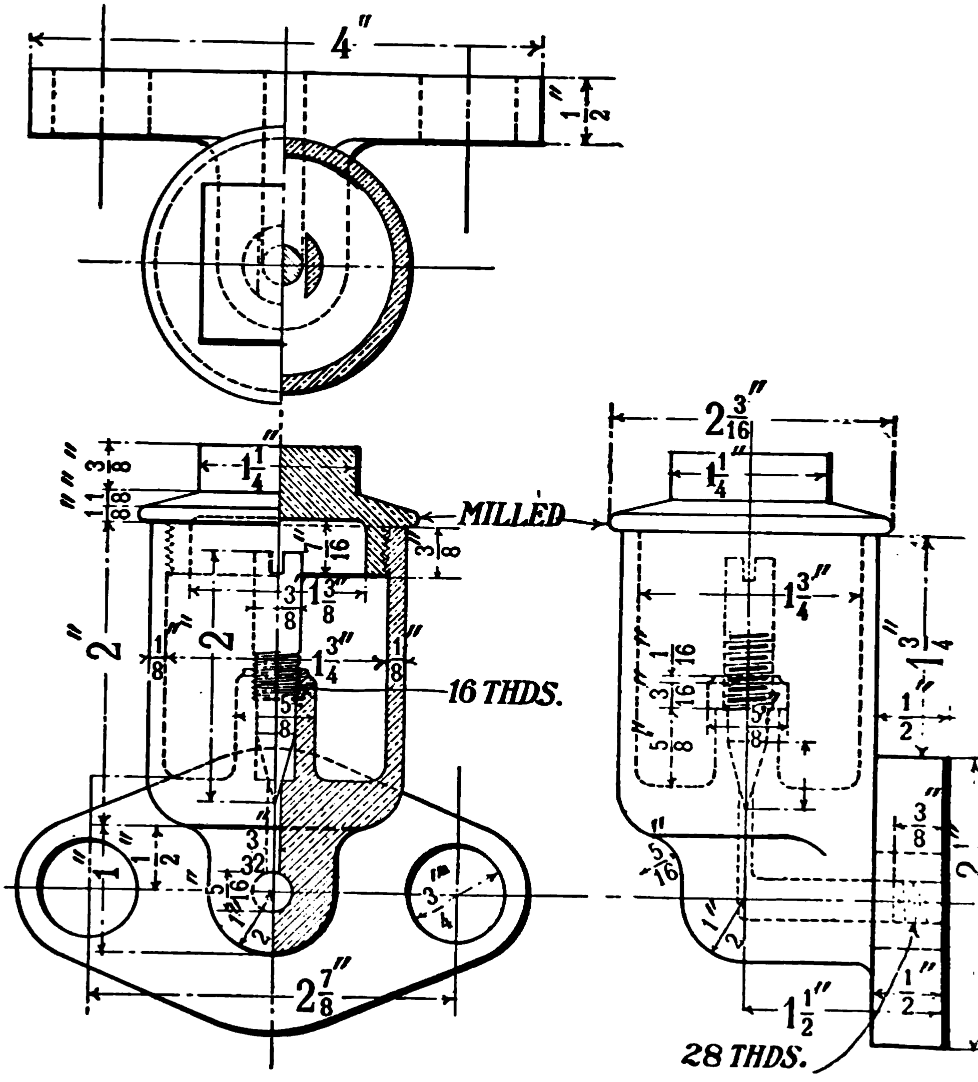

4"
1/2"
1 1/4"
3/8"
1/8"
1/8"
MILLED
7/16"
3/8"
3/8
1 3/8"
2"
2"
1/8"
1 3/4"
1/8"
16 THDS.
5/8
1"
1/2"
3/32
5/16
1/2"
3/4"
2 7/8"
2 3/16"
1 1/4"
1 3/4"
1/16
3/16
5/8
5/8
1 3/4"
1/2
5/16"
3/8"
1/2"
2"
1 1/2"
28 THDS.

FIG. 224.

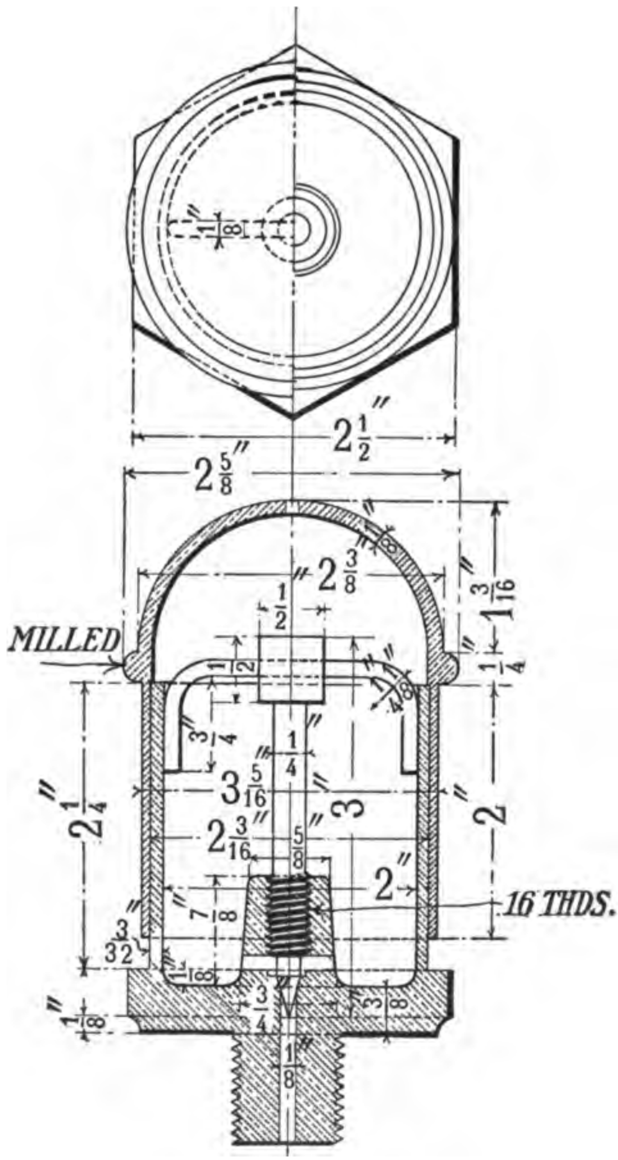


FIG. 225.

main rod on cross-head. It will be seen that in this case the flow of the oil is also mechanically controlled.

Fig. 225 is a form of cup used on the guides. The flow of the oil is in this case also regulated by the raising or the lowering of the spindle by hand.

Exercise 109.—Make drawings, as directed by the instructor, of one or more of the oil-cups illustrated in Figs. 220 to 225 when it is desired to fill unoccupied space on drawing-paper. *Scale, full size.*

CHAPTER X.

ENGINE DETAILS.

The Plain Slide-valve.—The construction of all slide-valves must be such as to satisfactorily meet the following requirements:

1. To admit steam to one end only of the cylinder at a time;

2. To allow the steam in the cylinder to escape from one end at least as soon as steam is admitted at the other end;

3. To prevent steam from entering the exhaust-port from the steam-chest.

During one revolution of the crank there are four principal points reached and passed by the valve in the course of its travel:

1. The *point of admission*, when steam begins to enter the cylinder. (See Fig. 236, Plate I.)

2. The *point of cut-off*, when steam is prevented from entering the cylinder. (See Fig. 233, Plate I.)

3. The *point of exhaust*, when steam is released from the cylinder. (Fig. 235, Plate I.)

4. The *point of compression*, when the exhaust is closed. (Fig. 234, Plate I.)

In Fig. 226 is given a longitudinal section of a plane slide-valve, and also of the valve-seat S of the cylinder. The valve is shown in its central position, X and Y are the steam-ports and Z the exhaust-port. The valve-face is the under side of the valve with a length equal to $F = 7\frac{1}{4}'' + 2L$.

Outside Lap, or simply *lap* is the darkened portion L of

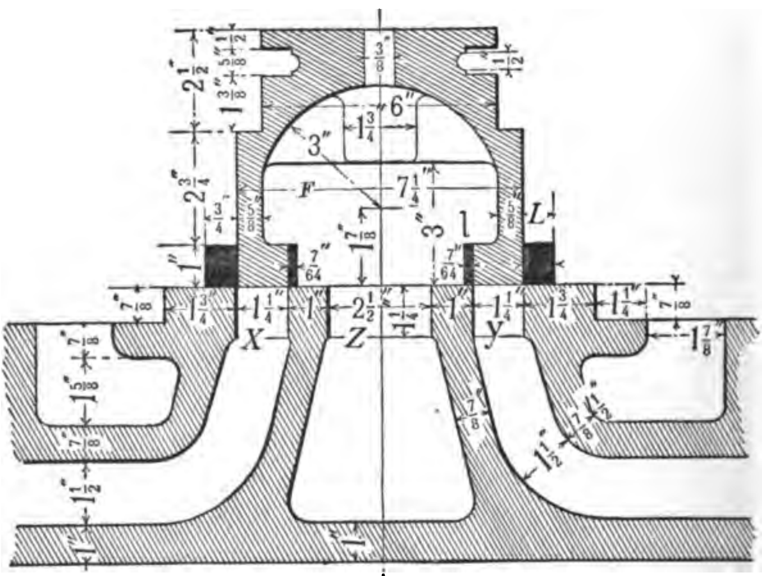


FIG. 226.

the valve which overlaps the steam-port when the valve is in its central position. Lap has no effect on compression or exhaust, but it hastens the cut-off, prolongs expansion, and shortens the time the port is open.

Inside Lap, the smaller darkened portion l which overlaps the bridge between the steam- and exhaust-ports, prolongs expansion, hastens and increases compression, retards

the exhaust, but does not affect the admission or point of cut-off.

The Travel of the valve is equal to twice the total distance it moves from its central position in either direction; or if the arms of the rocker are of equal lengths, then the travel of the valve is equal to twice the eccentricity of the eccentric. (See "Eccentric and Straps.") It is also equal to twice the sum of the width of the steam-port and lap plus the over-travel if any.

The Lead Angle is the angle made by the centre-line of the crank with the centre-line of motion of the engine when the crank is at the point of admission. (See Fig. 236, Plate I.)

The Lead is the amount which the valve has opened the steam-port at the beginning of the stroke. (See Fig. 231, Plate I.) To obtain smooth running, increased speed should have increased lead, and when the lead is increased every operation of the valve is quickened.

The Angle of Advance of the eccentric is the number of degrees which the centre-line of the eccentric is over 90° ahead of the centre-line of the crank without a rocker, and with a rocker it is the number of degrees short of 90° behind the crank. The first case is illustrated in the diagram in Plate II, as follows: Let *AO* be the centre of the crank, *CO* a line 90° ahead of it, and *OE* the centre-line of the eccentric. Then *COE* is the angle over 90° ahead of the crank, and is therefore the angle of advance. For the case with a rocker let *AO* be the centre of the crank as before, and *OD* a line 90° behind it, and *OF* the centre-line of the eccentric. Then the angle *DOF* is the angle short of 90° behind the crank, and is therefore the angle of advance.

Inside Clearance is the opposite of inside lap, instead of the valve overlapping the bridge when on the centre; as shown at *l* in Fig. 226, it shows a clearance between the inside edge of the valve and the bridge. Inside clearance hastens exhaust, delays compression, but has no effect on the cut-off or admission.

Overtravel is the distance the steam edge of the valve travels after fully opening the port, as shown in Fig. 232, Plate I. It increases the sharpness of the cut-off, retards compression, and gives a later release.

Cylinder Clearance is all that space between the faces of the piston and the valve when the piston is at the beginning of the stroke.

Piston Clearance is the distance between the piston and the cylinder-head. This clearance is to prevent the piston from striking either cylinder-head when the brasses on the connecting-rod wear and cause lost motion.

Point of Cut-off is the point on the crank-circle which the centre of the crank reaches when the valve cuts off the live steam from the cylinder, and for the remainder of the stroke utilizes the expansive power of the steam. (See Fig. 233, Plate I.)

Compression of the steam follows the closing of the exhaust before the piston has completed its stroke. This is done to obtain a yielding cushion for the reciprocating parts to come to a full stop without shock before beginning the return stroke.

Expansion begins at the point of cut-off and continues to the point of exhaust. (See Figs. 233 to 235 in Plate I.)

During this period the valve travels a distance equal to the outside lap plus the inside lap.

The Allen-Richardson Balance-valve.—This is one of the most popular combination slide-valves and is used on locomotives, stationary and large marine engines. Fig. 227 clearly shows the different parts used in the construction of this valve. The balance is effected by means of four rect-

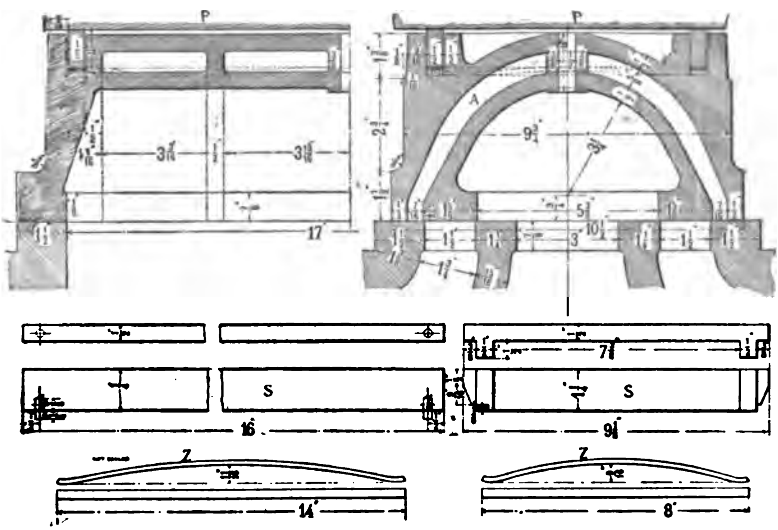


FIG. 227.

angular packing-strips *S* fitted into grooves on the top of the valve. Semi-elliptic springs *Z* are used to hold the packing-strips against the pressure-plate *P* when there is no steam in the chest, but when steam is admitted to the chest it forces the strips against the pressure-plate and sides of the grooves, forming a steam-tight joint and preventing the steam from acting on that part of the top of the valve enclosed by the four packing-strips.

Exercise 110.—Make drawings as shown in Fig. 227, and also a half plan of the top. Scale 8″ = 1 foot.

The Allen feature of this valve is the supplementary port shown at *A* just above the exhaust-arch. By means of this additional port steam is admitted to the same steam-port in the cylinder from both sides of the valve at the same time, thereby increasing the steam-supply with short cut-offs. The advantages of this valve over the plain slide-valve and the objections to it are discussed in the proceedings of the following societies: A. S. M. E., vol. 20, May 1899; The Western Railway Club, March, 1897; Am. Railway M. M. Association, 1896; and in the "Locomotive up to Date" by Chas. McShane.

* **The American Balance Slide-valve.**—The American Balance is applied to any type of slide-valve. It consists of a steam-tight joint being formed between the valve and the under side of the steam-chest cover, thus excluding live-steam pressure from a given area. (See Fig. 228.) This joint is formed by a bevelled snap-ring which, when in place, is slightly expanded over a cone. The cone or cones are either cast with the valves or bolted to it, as circumstances require.

The mechanical construction of the balance is: First, the cone or two cones, where necessity requires, are either bolted to or cast with the valve. The snap-rings, which are bevelled on their inner side to a corresponding degree with that of the cone, are bored smaller in diameter than their required working diameter so that, by their being forced down on the cone by the placing of the steam-chest cover in position, the rings

* The above description was furnished by Mr. J. T. Wilson, General Manager of The American Balance Slide-valve Co.

themselves are under tension and are thus supported by their own elasticity when not under steam. The steam when admitted to the steam-chest exerts a pressure on the entire circumference of the ring, which has a tendency to close it or decrease its diameter, and owing to its bevelled face and the taper of the cone the steam also acts to lift it. By careful

FIG. 228.

consideration of the operation of this ring, now being held by the steam-pressure tightly against the face of the cone, it will at once be seen that all lateral wear is avoided, and the ring moves as a part of the cone or valve itself. It will also be noted that the ring is absolutely compelled to assume its working position by the pressure on its circumference. When steam is shut off from the engine and the engine allowed to drift, as in locomotives, the valve is free to leave its seat until

the cone comes in contact with the cover. This affords perfect and ample relief of the air which the piston is forcing from one end of the cylinder, and also a direct communication with the other end of the cylinder, in which a vacuum is being formed. The cylinders are therefore perfectly relieved by allowing the valve to lift $\frac{1}{8}''$ off its seat.

The bevelled feature in the ring renders the ring self-sup-

FIG. 229.

porting when not under steam, and supported by the steam-pressure when under steam, automatic adjustment for the wear, positive action under all conditions, and self-maintaining the steam-joint. It renders it possible also to duplicate the rings of respective sizes in repairs.

Owing to the absence of lateral wear on the cones new rings can be duplicated at any future time. The greatest area of balance can be secured by this design, because it is least affected by back or upward pressure. The valve in order to

leave its seat must first expand the taper ring against the chest-pressure acting on its circumference.

The features enumerated all depend upon the taper.

Fig. 228 is a double-cone balance-valve used on locomotives. The improved T ring, the invention of Mr. J. T. Wilson, is clearly shown in the figure.

Fig. 229 is a single-cone balance-valve for use on compound stationary engines. A double-cone valve of this kind is in use on the Japanese cruiser "Chtose," the rings of which

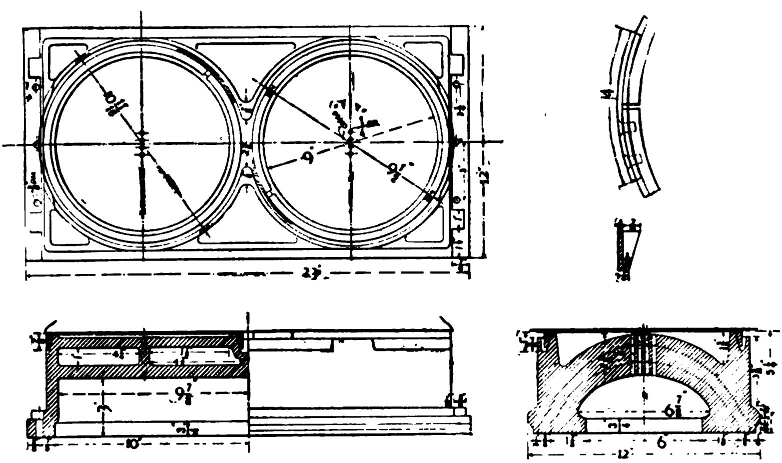


FIG. 230.

are three feet ten inches in diameter, while that in Fig. 229 is only twenty inches diameter.

Exercise 111.—Make drawings of Fig. 230 as shown. Scale 6″ = 1 foot.

The Bilgram Diagram.—Among the many diagrams devised to determine quickly and accurately the position of the valve for any position of the crank, that due to Mr. Hugo Bilgram is one of the simplest and best.

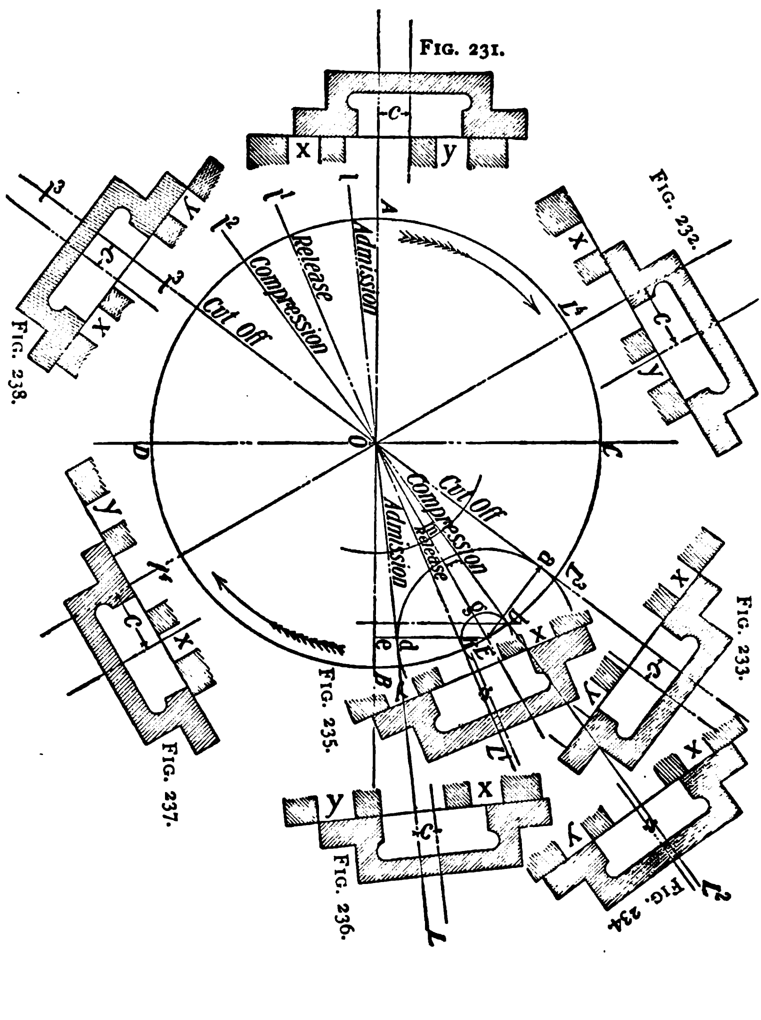


FIG. 231.
FIG. 232.
FIG. 233.
FIG. 234.
FIG. 235.
FIG. 236.
FIG. 237.
FIG. 238.
Admission
Release
Compression
Cut Off
Cut Off
Compression
Release
Admission
A
B
C
D
O

PLATE I.

In Plate I let AB represent the valve circle, equal in diameter to the travel of the valve, and Ll the centre-line of the crank rotating in the direction of the arrow. From B lay off the angle EOB, equal to the angle of advance. At E describe the arc bgk with a radius equal to the inside lap, and also the arc afd with a radius equal to the outside lap. Crank positions drawn tangent to these arcs at a, b, k, and d will give the points of cut-off, compression, release, and admission respectively, as indicated in the figure.

Let us follow the crank through one revolution, beginning with the dead-point A. In this position de is equal to the outside lead, and the valve has moved from its central position a distance Ee equal to the lap plus the lead. These relations are clearly shown in Fig. 231. c gives the distance which the valve has travelled from its central position, and at X the left-hand steam-port is shown open to steam an amount equal to the lead when the piston is at the beginning of its forward stroke, and the eccentric is connected directly to the valve, i.e., without a rocker.

When the crank reaches the position L^1 perpendicular to OE the valve will have travelled from its central position a distance equal to EO. This is the extreme position of its forward travel, as shown in Fig. 232. The maximum opening of the port X to steam is equal to Of, and the overtravel to mf, the actual width of the steam-port being $= Om$.

As the crank leaves L^1 the valve begins to return, and when the crank is at L^2 the distance of the valve from its central position is equal to the lap ab. Port X is now closed to steam, and cut-off is accomplished as shown in Fig. 233.

When the crank is at L^3 the right-hand steam-port is

closed to exhaust, and compression begins as shown at *Y*, Fig. 234.

When the crank reaches L^1 the valve is on the point of opening port *X*, Fig. 235, to release the steam which was under compression during the time the crank moved from L^2 to L^1. At crank position *L* we find, as shown in Fig 236, that the valve is on the point of admitting steam to port *Y*, and at *B* the backward stroke of the piston begins, the valve having opened the port an amount equal to the lead *de*, equal to the opening shown at *X* in Fig. 231.

At crank position l^1 and valve position Fig. 237 the valve has attained its maximum travel in the opposite direction to that shown in Fig. 232. At l^2, Fig. 238, the valve cuts off steam from port *Y*, and at *A* the *new* forward stroke begins.

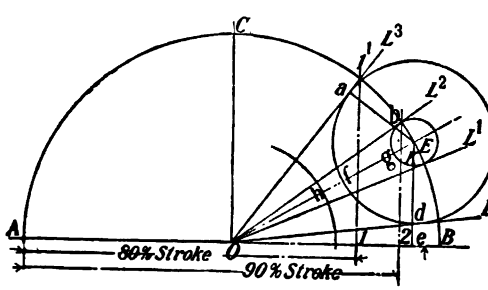


FIG. 239.

Exercise 112.—(Fig. 239.)

Given.			Required
Travel	=	5″.	Outside lap.
Angle of advance...	=	30°.	Inside lap.
Cut-off............	=	80% of stroke.	Outside lead.
Compression.......	=	90% of stroke.	Inside lead.
Width of steam-port	=	1¼″.	Maximum port opening.
			Overtravel.

Draw AB and CO at right angles to one another. Describe the valve-circle arc ACB with a radius equal to half the travel or eccentricity of the eccentric $= 2\frac{1}{8}''$ to the scale of twice full size. From B lay off the angle EOB equal to the angle of advance $= 30°$. Let AB represent the stroke, and from A lay off $Al =$ 80% of a stroke of 24″, and erect a perpendicular to cut the valve circle in l^1. Draw OL^1 through l^1; this is the crank position at the point of cut-off. Through E perpendicular to OL^1 draw Ea. With centre E and radius Ea describe the lap circle *afd*. From A lay off $A2 =$ 90% of the stroke, and erect a perpendicular to cut the valve circle at b. Through b draw OL^2, which is the crank position at the point of compression. With E as centre and Eb as radius describe the inside lap circle. Draw OL^1 tangent to *bgk* at point k.

At O with a radius = the port opening describe the arc h. Then Ea is the required lap, Eb the inside lap, *de* the lead, *ke* the inside lead, *Of* the maximum port-opening, and *hf* the overtravel.

Exercise 113.—(Fig. 240.)

Given.		Required.
Cut-off	= 80% of stroke.	Travel of the valve.
Lap	= 1″.	Angle of advance.
Lead	= $\frac{1}{4}''$.	

Draw to a scale equal to twice full size AB and CO at right angles. Draw OL^1, the position of crank at cut-off. Draw line 1 2 parallel to AB at a distance above it equal to the lead *ed*. Draw line 3 4 parallel to AB at a distance equal to the lap plus the lead above it. With a radius equal to the

given lap find by trial a centre on the line 3 4, and draw the lap circle *afd* tangent to OL^3, and line 1 2 at the points *a* and *d*.

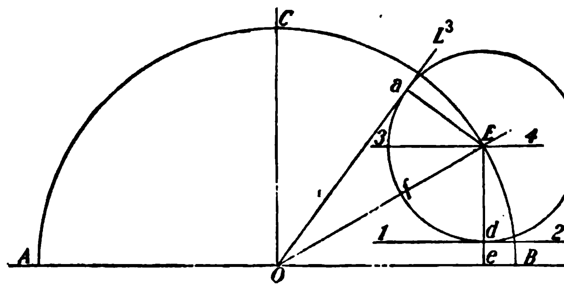


FIG. 240.

Then through *E* with centre *O* describe the valve circle *ACB*.

AB is the *travel* of the valve, and *EOB* the *angle of advance.*

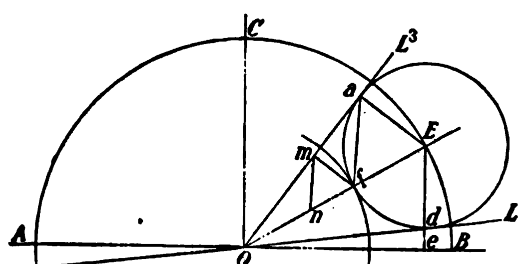


FIG. 241.

Exercise 114.—(Fig. 241.)

Given.		Required.
Cut-off............	= 80% of the stroke.	Travel of the valve.
Admission	= 90% of the stroke.	Lead.
Maximum port-opening	= *Of* (Fig. 240).	Angle of Advance.
		Lap.

Draw AB and CO at right angles. Draw OL, the position of the crank at the point of admission. Draw OL', the crank position at cut-off, and arc f with a radius equal to the given maximum port-opening. Bisect the angle LOL' with the line OE. The centre of the lap-circle will be on this line. Draw fm perpendicular to OL', and make $fn = fm$. Through f draw fa parallel to nm, and aE parallel to mf. E is the centre of *lap circle*. Through E describe the valve circle ACB, and draw the line Ee at right angles to AB.

Then AB is the *travel of the valve*, Ea the lap, BOE the *angle of advance*, and de the lead.

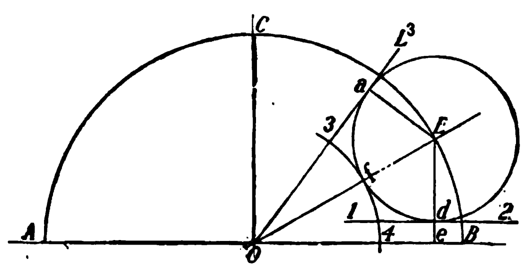


FIG. 242.

Exercise 115.—(Fig. 242.)

Given.		Required.
Cut-off.............	= 80% of the stroke.	Angle of advance.
Lead...............	= $\frac{1}{4}''$.	Lap.
Maximum port-opening	= Of (Fig. 240).	Travel of valve.

Draw AB and CO at right angles. Locate the crank position OL'. Draw the lead-line 1 2 at a distance de from AB and parallel to it. With centre O describe arc 3 4 with a radius equal to the maximum port-opening. Find by trial the centre E of a circle that can be drawn tangent to OL', arc 3 4, and line 1 2. Through this centre draw OE.

Then *BOE* is the angle of advance, *Ea* the lap, and twice *OE* is equal to the travel of the valve.

The Zeuner Valve Diagram.—In Plate II let *AB* represent the stroke of the piston, the circle *ACBD* the path of the crank-pin, and *L* the centre-line of the crank.

From *C* lay off *OE* equal to the angle of advance, and on *OE* as a diameter describe the valve circle equal to half the travel of the valve or eccentricity of the eccentric when no rocker is used. From centre *O* draw the arcs *abc* and *gfh* equal to the outside and inside laps respectively.

At the beginning of the forward stroke the true position of the crank would coincide with *AO*, and the centre-line of the eccentric with *OE*.

Now since the position of the point *E* is fixed for a given eccentricity and angle of advance, the point *E* will always be found on the circumference of the circle having *OE* as a diameter; and if the valve circle together with the crank be rotated around the centre *O* in a direction opposite to the arrow, its intersection with the line *OB* from *O* will be the distance which the valve has travelled from its central position after the crank has moved through any given angle.

But instead of rotating the crank and valve circle let them remain fixed and rotate the line *OB* as an imaginary crank in the direction of the arrow, and the same results will be obtained in a much simpler way.

Draw *OL*, the imaginary crank, through the point where the lap arc *abc* intersects the valve circle at the point *c*. The position of the valve will then be at the point of admission, because the valve will have travelled from its central position a distance equal to *Oc*, equal to the lap.

PLATE II.

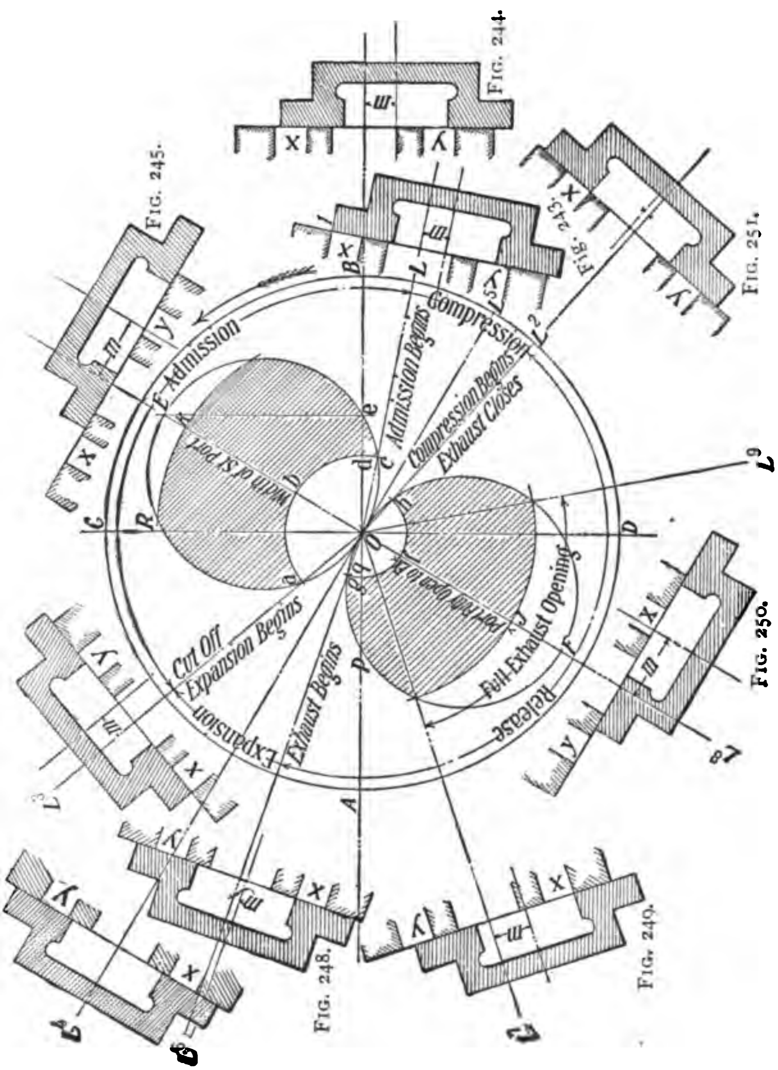


FIG. 243.
FIG. 244.
FIG. 245.
FIG. 246.
FIG. 247.
FIG. 248.
FIG. 249.
FIG. 250.
FIG. 251.
Admission
Admission Begins
Compression
Compression Begins
Exhaust Closes
Width of St. Port
Cut Off
Expansion Begins
Expansion
Exhaust Begins
Full Exhaust Opening
Release
Port full open to Ex.

This is clearly shown in Fig. 243. The valve is travelling in a direction opposite to the imaginary crank, and the steam edge l of the valve is on the point of admitting steam to the port X just before the beginning of the forward stroke. When the imaginary crank reaches the position OB the valve will have travelled a distance equal to Oc from its central position and, OB being a dead-centre, the valve will have opened the port X to steam an amount equal to the lead, and the port Y to exhaust an amount equal to pq, Fig. 244.

When the crank has reached the position OE the valve will then have attained the extreme position of its travel. The shaded part bk shows the full opening of the steam-port, and ke the amount of overtravel, and at the same time the port Y is fully open to exhaust, as shown by the shaded portion fJ, and JF is the exhaust overtravel (Fig. 245).

The valve now returns and at R begins to close port X, until when the crank arrives at L^3 the port is fully closed and cut-off takes place, as shown in Fig. 246.

When the crank is in the position L^4 at right angles to OE the valve is in its middle position, as shown in Fig. 247.

At the crank position L^5 the valve has travelled a distance Og from its central position, and the port X is about to open to exhaust, as shown in Fig. 248. The port continues to open, until at the position L^7 the port is fully open and continues so until the crank reaches the position L^8, when it begins to close, and is fully closed when the crank reaches L^9. Now compression begins and continues through the angle L^9OL. At L the valve has returned to the point of admission a little before the beginning of the new forward stroke.

At crank position L^6 it will be seen from Fig. 250 that the

port *Y* is fully open to steam, the port *X* fully open to exhaust, and that the valve has reached the extreme position of its travel for the backward stroke—just the opposite of the position shown in Fig. 245.

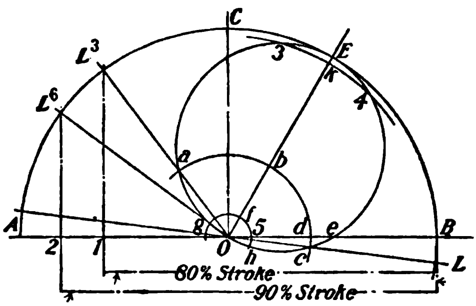


FIG. 252.

Exercise 116.—(Fig. 252). Assume the same conditions as in Ex. 112 for the Bilgram diagram. Draw *AB* and *CO* at right angles. Make *AB* to any convenient scale equal to the stroke of the piston, and let *ACB* represent the path of the crank-pin. From *C* lay off angle *OE* equal to the angle of advance = 30°, with a scale equal to twice full size. On *Ok* as a diameter, equal to half the travel of the valve, or $2\frac{1}{2}''$, describe the valve circle *Oakc*. From *B* lay off *Bl* equal to 80% of the stroke, and erect a perpendicular to cut the crank-pin arc in L^3. Draw OL^3, the position of the crank at cut-off. Through the point *a*, where OL^3 cuts the valve circle, with *O* as centre describe the arc *abc*. From *B* lay off *B*2 equal to 90% of the stroke, and erect a perpendicular to L^6. Draw OL^6, and through the point *g*, where OL^6 intersects the valve circle, with *O* as centre describe the arc *gfh*. From *b* lay off *bk* equal to the width of the steam-port, and with centre *O* and radius *Ok* describe the arc 3*k*4.

Then *Oa* is the required lap, *Og* the inside lap, *de* the lead, 5*e* the inside or exhaust lead, *OE* the maximum port-opening, and *KE* the overtravel.

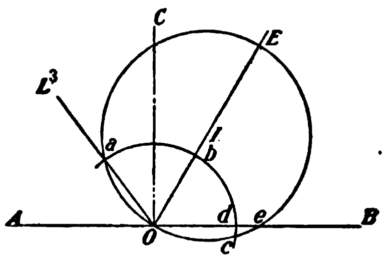


FIG. 253.

Exercise 117.—(Fig. 253.) Assume the same conditions as given in Ex. 113.

Draw *AB* and *CO* at right angles. Draw OL^3, the crank position at cut-off. From *O* describe arc *abc* with a radius equal to the lap, scale as before. Lay off *de* equal to the lead. Bisect *Oa* and *Oc*, and the point *I* where the bisectors intersect will be the centre of the valve circle which may now be drawn through the points *aOe*.

Then *OE* is equal to half the travel of the valve, and *COE* is the angle of advance.

Exercise 118.—(Fig. 254.)

Given.		Required.
Point of cut-off...	= 80% of stroke.	Travel of valve.
Point of admission	= 90% of stroke.	Lap.
Lead.............	= $\frac{1}{4}''$.	Angle of advance.

Draw *AB* and *CO* at right angles. Draw the crank positions *OL* and OL'. Bisect the angle LOL' with the line *OE*. On *OE* assume any point as *g*, and draw *gf* perpendicular to *OB*, and *gh* perpendicular to *OL*. With center *O* and radius *Oh* describe the arc *he*.

Now the angles gOB and gOL are constant for a given admission and cut-off; therefore the lead will vary directly as the eccentricity.

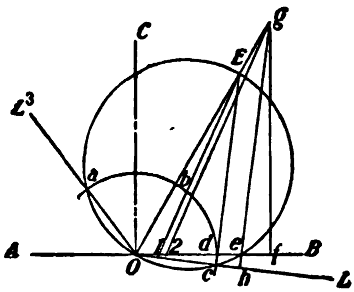


FIG. 254.

Let Og be an assumed eccentricity, then ef will be its corresponding lead, and the given lead is to the assumed lead ef as the required eccentricity is to the assumed eccentricity Og.

Lay off $O1$ equal to the given lead, and $O2$ equal to ef. Draw $2g$, and $1E$ parallel to $2g$.

With a radius equal to Oe minus the given lead and centre O describe the arc abc. On OE as a diameter describe the valve circle $Oaec$.

Then COE is the required angle of advance, OE the eccentricity or half the travel of the valve, and Oa the lap.

Exercise 119.—(Fig. 255.) Assume the conditions as in Ex. 115.

Draw AB and CE at right angles to each other. Draw OL', the crank position at cut-off, Oa, the given lead, and Ob, the given maximum port-opening. At b erect the perpendicular bg. Through a draw ad at right angles to OL'. Bisect the angle dce with the line Oc, and produce it to intersect fg at g. Join

ag, and draw *cf* parallel to *Ob*. With *c* as centre and *cf* as radius describe arc *fk*, cutting *ag* in *k*. Join *ck*, and draw *a O* parallel to *kc*, cutting O_1g in the point O_1. Draw O_1e parallel to O_1A.

Then *Ock* is the required angle of advance, O_1a half the required travel of the valve, and O_1e the lap.

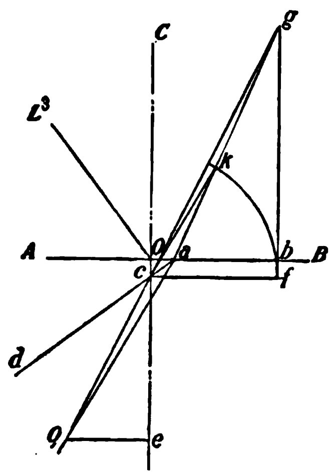


FIG. 255.

Engine Frame or Bed-plate.—Frames for horizontal engines are usually made of cast iron. This is the most suitable material owing to the complicated sections found in most frames, and also because it gives the necessary rigidity.

Figs. 256 to 258 show an engine frame of the "Tangye" type. It is that used by the Buckeye Engine Co. of Salem, Mass.

Exercise 120.—Make drawings as shown in Figs. 256 to 258. Scale $1\frac{1}{2}'' = 1$ foot.

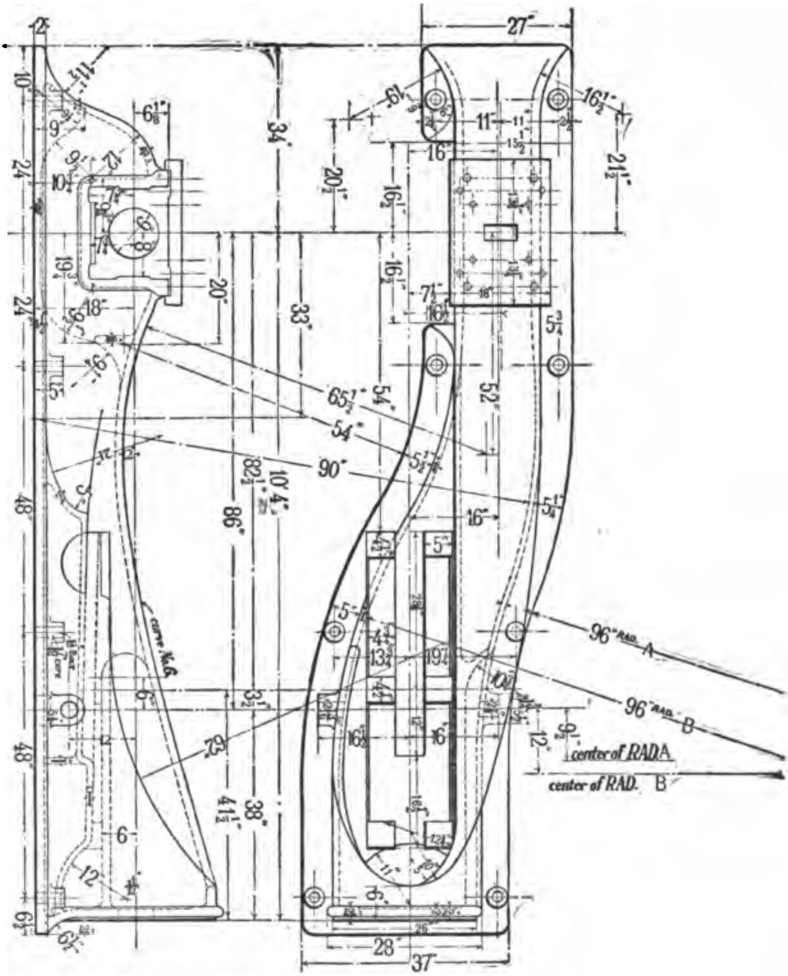


FIG. 256.

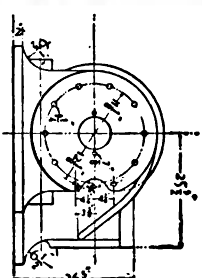


FIG. 258.

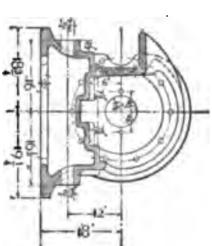


FIG. 257.

Cylinder.—Steam-engine cylinders are almost always made of a tough, close-grained cast iron as hard as can be safely worked.

Diameter of Cylinder, D.

Let P = the mean effective pressure of steam in pounds per square inch = M. E. P.;
L = length of stroke in feet;
A = area of piston in square inches;
N = number of strokes per minute.

Then $\frac{PLAN}{33000}$ = I.H.P. or indicated horse-power, and

A is therefore $= \frac{\text{I.H.P.} \times 33000}{PLN}$; so $D = \frac{A}{.7854}$.

The mean effective pressure P may be found from the following formula:

Let p = the absolute initial pressure of steam, i.e., the gauge-pressure + 15 lbs.; − 7% for loss between boiler and cylinder.
r = ratio of expansion = length of stroke in inches ÷ distance travelled by piston in inches before steam is cut off.

Then $P = \frac{1 + \text{hyp. log. } r}{r}$ − back-pressure.

Thickness of Cylinder, t.

Let P_1 = boiler-pressure of steam per square inch in pounds;
D = diameter of cylinder in inches.

"Whitham" recommends the following formula for horizontal or vertical cylinders of large or small diameter where

provision is made for reboring and sufficient strength and rigidity are secured:

$$t = 0.03\sqrt{P_1D}.$$

Length of Cylinder.—The length of cylinder between heads = Stroke + thickness of piston + the sum of the piston clearance at both ends.

Cylinder Head.—The cylinder head or cover next to the crank is sometimes cast on the cylinder.

The thickness of the cylinder head recommended by Prof. Seaton is

$$\frac{P_1D + 500}{2000}.$$

The thickness at the flange where the head is bolted to the cylinder should be $\frac{1}{8}$ greater than this.

Cylinder-head Bolts.—The diameter of cylinder-head studs in locomotives is usually $\frac{7}{8}''$, and their pitch about 4 times their diameter.

For stationary and marine practice "Ripper" gives

$$n\frac{\pi}{4}d^2f = \frac{\pi}{4}D^2p,$$

where n = number of studs;
d = diameter of studs;
D = diameter of cylinder;
p = maximum pressure of steam;
f = 4000 to 5000.

Steam-ports.—The steam-ports which conduct the steam from the valve-chest to the cylinder should be as short and direct as possible, but large enough to prevent wire-drawing and of easy curvature.

The length of ports in locomotives is usually 1″ less than the diameter of the cylinder.

In other types the length is generally made $0.8D$.

The area of the steam-port is given by many authorities as follows:

$$a = \frac{AV}{v},$$

where A = area of the piston in square inches;

v = velocity of steam through the port = 6000 feet per minute;

V = velocity of piston in feet per minute (from 1000 to 1200);

a = area of steam-port.

Figs. 259 to 262 show the working drawings of a horizontal steam-engine cylinder made by the Buckeye Engine Co. Fig. 259 shows a longitudinal section through the centre of the cylinder, Fig. 260 a cross-section through the exhaust-passage, Fig. 261 a back-end view showing the opening for a valve-rod, and Fig. 262 a plan.

Figs. 263 and 264 are the heads and covers suitable for this cylinder. When in position on the engine-frame the end of the cylinder, shown in Fig. 261, is bolted to the end of the frame shown in Fig. 261.

Exercise 121.—Make drawings of steam-cylinder as shown in Figs. 259 to 262. (*Scale 1½″ = 1 foot.*)

Exercise 122.—Make the design of a cylinder similar to that shown in Figs. 259 to 262 to develop 100 I.H.P. Stroke 30″; steam-pressure 90 lbs. per square inch; cut-off at 50% of the stroke. Number of strokes per minute 220.

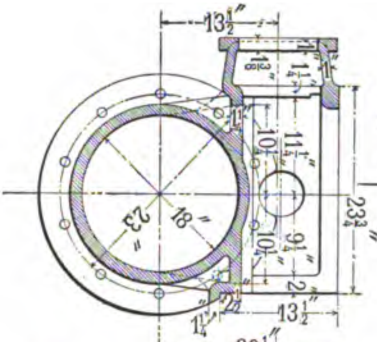


FIG. 260.

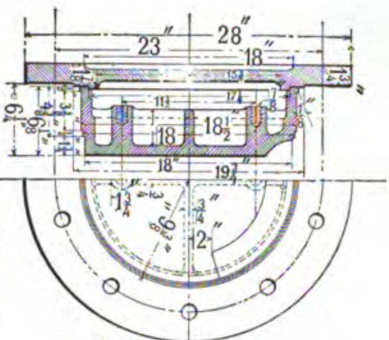


FIG. 263.

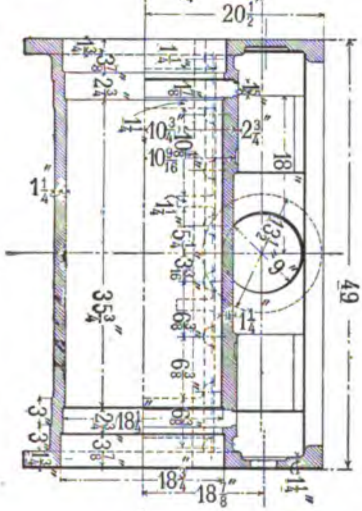


FIG. 259.

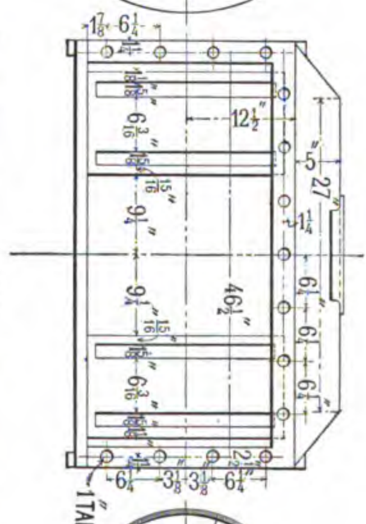


FIG. 262.

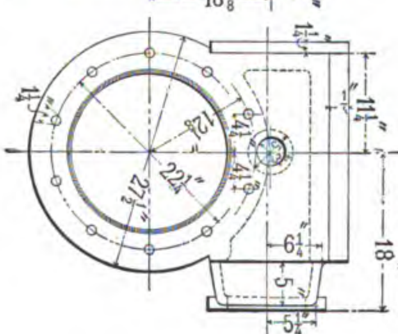


FIG. 261

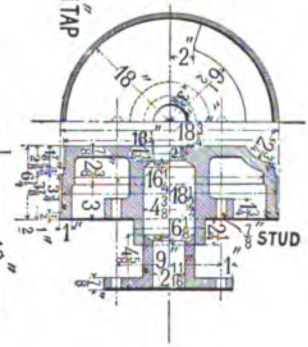


FIG. 264.

Exercise 123.—Make drawings of the cylinder heads shown in Figs. 263 and 264. (*Scale 1½" = 1 foot.*)

Pistons.—A piston is that part of an engine or pump which slides to and fro inside a hollow cylinder either driven by fluid pressure or acting against fluid pressure they are usually of circular section and are made of brass, wrought iron, cast iron, or steel.

A piston with valves which permit the fluid to pass from one side to the other is called a *bucket* and is used in pump cylinders.

A single-acting piston, guided by the stuffing-box instead of the cylinder, is called a *plunger* and is also used in pumps.

Steam-pistons.— A steam-piston should be designed so as to prevent the steam from passing from one side of the piston to the other.

The spring packing-rings should not press against the cylinder more than is necessary for steam-tightness.

A piston should be no heavier than is necessary for strength.

The weight of the piston should be distributed so as to prevent the excessive internal wear of the cylinder.

The piston must be firmly connected to the piston-rod.

Many different designs have been adopted to secure the above requirements.

Fig. 265 is a plain box piston, used by the Southwark Foundry & Machine Company. It is cast in one piece; the core being removed by three holes, shown in the front, which are afterwards plugged up. The two small holes are for eye-bolts which are used to remove the piston from the cylinder when necessary. The packing consists of two cast-iron spring

rings cut as shown in detail in the figure. The rod is forced into place by pressure and the ends riveted over.

Exercise 124.— Make drawings as shown in Fig. 265. (*Scale 6″ = 1 foot.*)

Fig. 266. This is another style of box pattern, used by the Ball Engine Company. The style of packing and the method of securing the piston-rod are plainly shown in the figure.

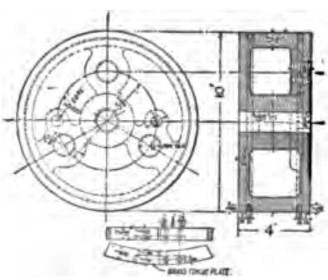


FIG. 265.

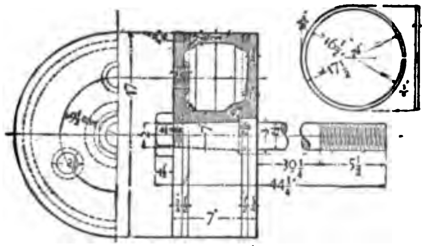


FIG. 266.

Exercise 125.—Make drawings as shown in Fig. 266, and in addition make a half end section through the centre of the piston. (*Scale 6″ = 1 foot.*)

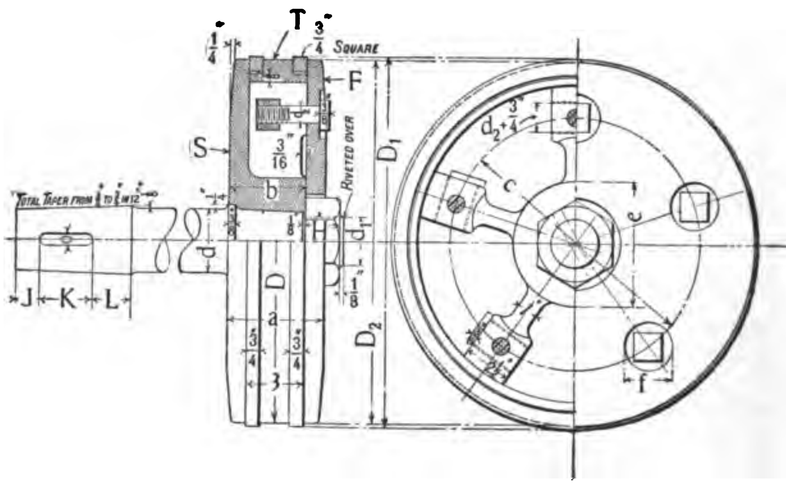
Fig. 267 shows a common built-up piston used largely in locomotives. It consists of a *spider*, *S*, a T ring, a follower, *F*, and two cast-iron spring-rings. The rod is forced into place and held by nut over which the end of the rod is riveted.

Exercise 126.—Make drawings of a built-up piston like Fig. 267 for an engine whose cylinder is 18″ × 24″. Take dimensions from Table 36. (*Scale 6″ = 1 foot.*)

Fig. 268, a cast-iron box piston used in the cylinder of the Empire State Express locomotive. Its construction is plainly shown in the figure.

Exercise 127.—Make drawings of the piston shown in Fig. 268. (*Scale 6″ = 1 foot.*)

Fig. 269 is a cast-steel box pattern cast in two parts and



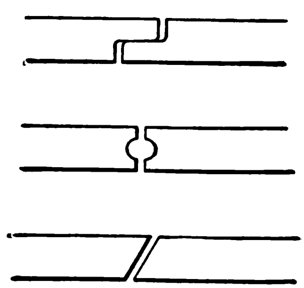


FIG. 267.

TABLE 86.

D	D_1	D_2	a	b	c	d	d_1	d_2	e	f	H	J	K	L	O
15	$15\frac{3}{16}$	$14\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{3}{4}$	10	$2\frac{1}{2}$	$1\frac{3}{4}$	$\frac{3}{4}$	$5\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$1\frac{3}{8}$	$\frac{9}{16}$
16	$16\frac{3}{16}$	$15\frac{7}{8}$	5	4	11	$2\frac{3}{4}$	2	$\frac{7}{8}$	6	$2\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$	$\frac{9}{16}$
18	$18\frac{3}{16}$	$17\frac{7}{8}$	5	4	13	$3\frac{1}{8}$	$2\frac{1}{4}$	$\frac{7}{8}$	$6\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$2\frac{5}{8}$	$1\frac{3}{4}$	$\frac{9}{16}$
19	$19\frac{1}{4}$	$18\frac{3}{4}$	5	4	14	$3\frac{1}{4}$	$2\frac{1}{2}$	$\frac{7}{8}$	$7\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$2\frac{3}{4}$	2	$\frac{5}{8}$
20	$20\frac{1}{4}$	$19\frac{3}{4}$	$5\frac{1}{4}$	$4\frac{1}{8}$	15	$3\frac{1}{2}$	$2\frac{3}{4}$	1	8	$2\frac{3}{4}$	2	$1\frac{1}{4}$	3	$2\frac{1}{4}$	$\frac{5}{8}$
22	$22\frac{5}{16}$	$21\frac{3}{4}$	$5\frac{1}{4}$	$4\frac{1}{8}$	17	$3\frac{3}{4}$	3	1	$8\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{4}$	3	$2\frac{1}{4}$	$\frac{11}{16}$

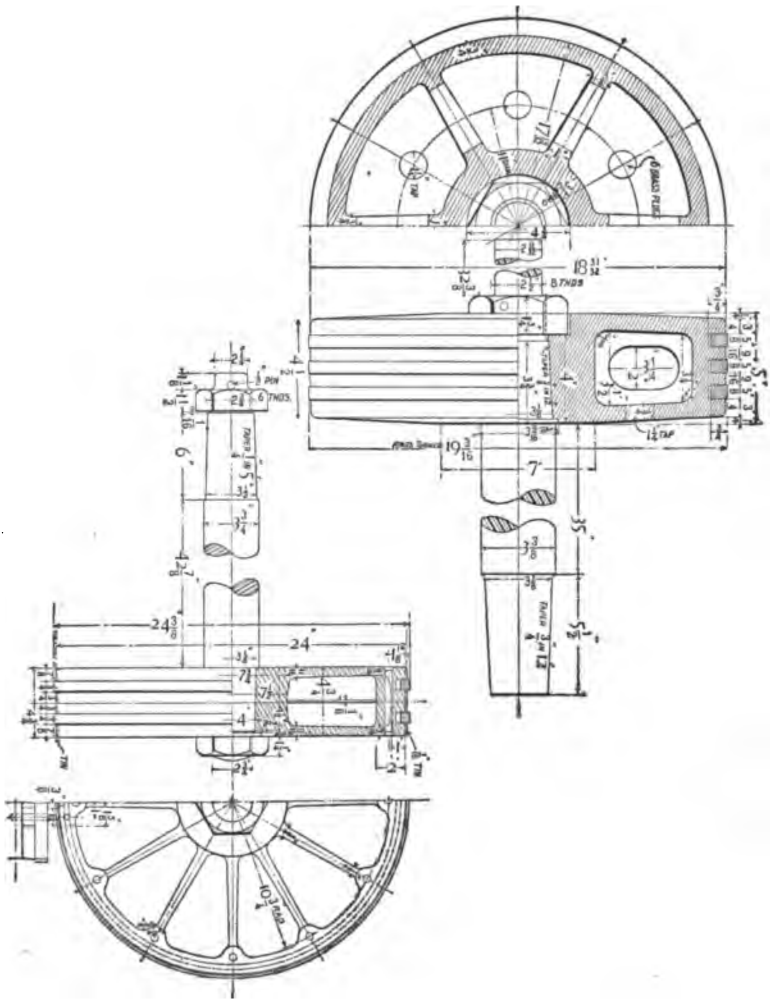


FIG. 268.

FIG. 269.

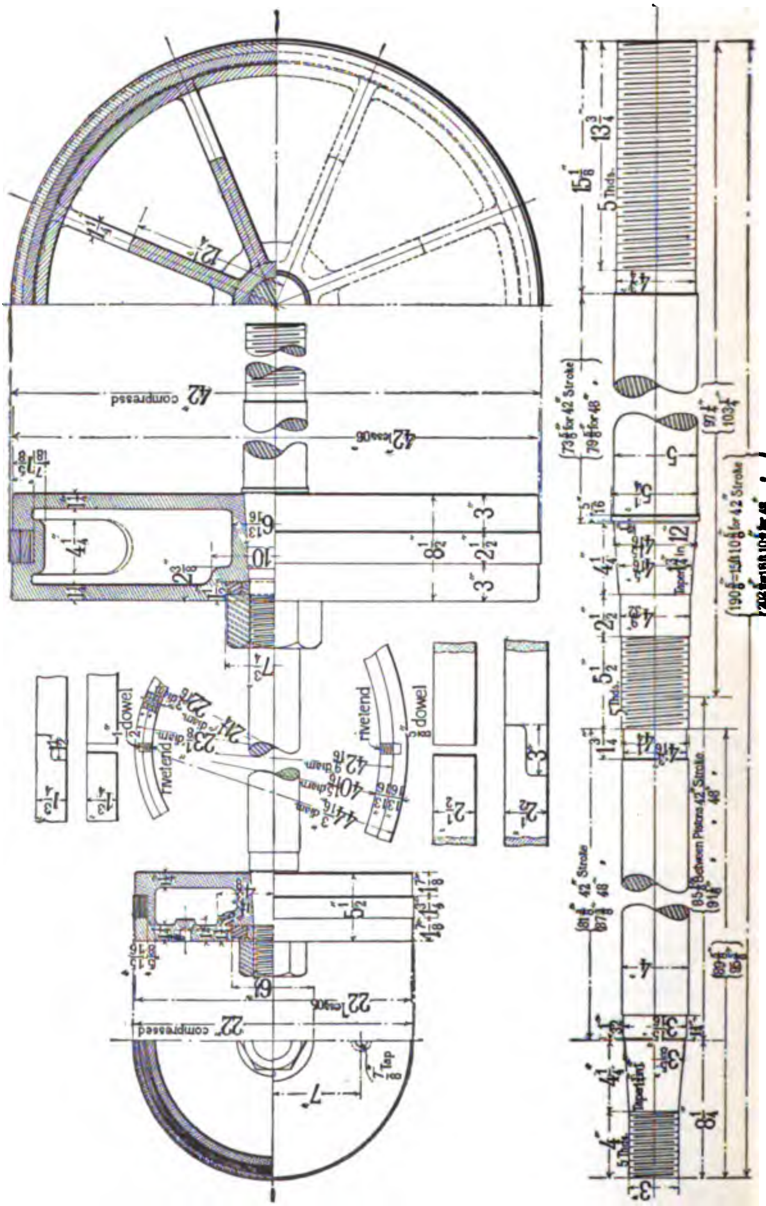


FIG. 270.

held together by rivets. This piston is made by the Baldwin Locomotive Works for the "Vauclain" compound locomotive.

Exercise 128.—Make drawings as shown in Fig. 269. (*Scale 4″ = 1 foot.*)

Fig. 270 shows the cast-iron pistons used in tandem stationary engines built by McIntosh & Seymour.

The packing is composed of cast-iron spring-rings cut and kept in place by the method shown in detail in the figure. The arrangement for securing the rod is shown in detail in Fig. 68, page 103.

Exercise 129.—Make drawings as shown in Fig. 270. (*Scale 3″ = 1 foot.*)

Fig. 271 is a built-up piston for the Tangye stationary engines made by the Buckeye Engine Co. It consists of a spider, follower, and adjusting-screws. There are no springs; the screws act on an uncut junk-ring, so can only be used for centring, not for packing. The packing-rings are turned larger than the bore of the cylinder so as to pack by their own elasticity. They may or may not be turned eccentric, that is, thin where cut, and full thickness opposite the cut.

If made eccentric, it is for the reason that they will be more nearly round when sprung into the cylinder.

Exercise 130.—Make drawings as shown in Fig. 271. (*Scale 6″ = 1 foot.*)

Fig. 272 shows a water-piston suitable for cylinders under 9″ diameter.

The piston-rod is fitted to the head with a shoulder to drive the piston, and the rod is secured in place by a nut. The follower is also held by a nut and lock-nut. By this

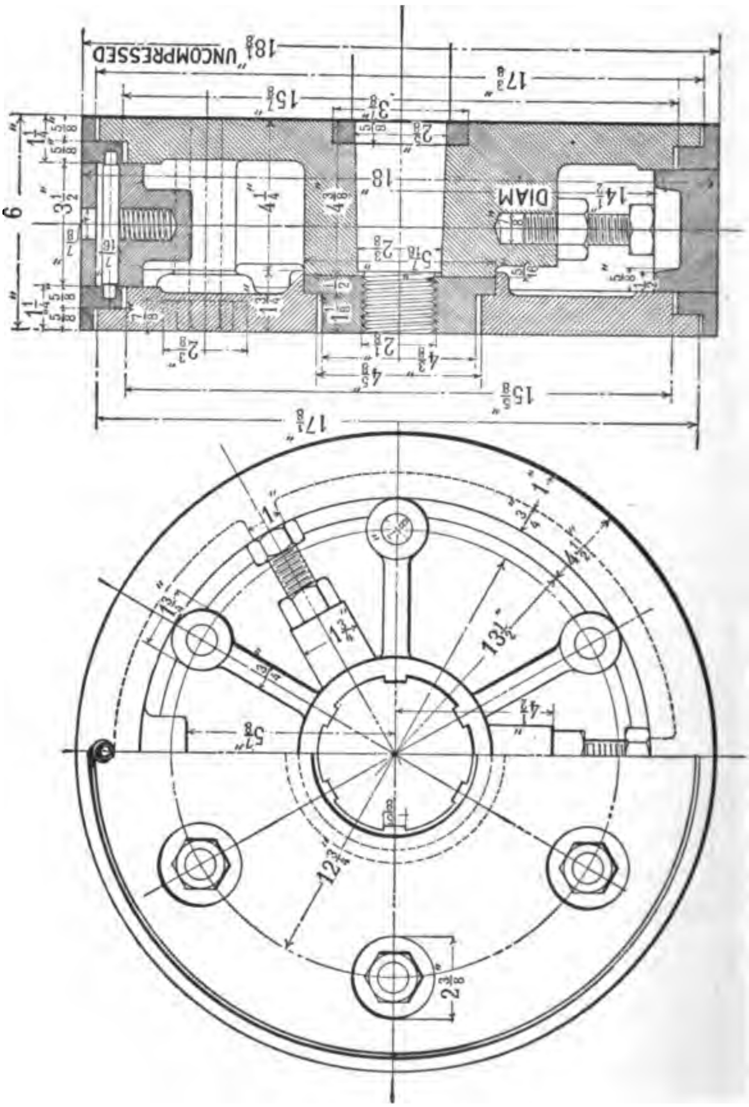


FIG. 271.

means the follower and packing may be adjusted or renewed at will. The packing is made of layers of cotton cloth and sheet rubber.

Exercise 131.—Make drawings of water-piston as shown in Fig. 272. (*Scale full size.*)

Connecting-rods.—In steam and other engines the connecting-rod connects the rotating crank with the reciprocating cross-head.

There are many styles of connecting-rods, and various methods are employed for taking up the wear of the brasses. Figs. 273 to 276 show good examples of rods used in stationary, locomotive, and marine engines of the most modern types.

Fig. 273 is the rod used by the Buckeye Engine Co. for their "Tangye" type of engine. The crank end is *solid*, the brasses are lined with babbitt, and adjustment for wear is had by means of a tapered steel block and screws. The cross-head end is called a *strap* end. The strap is firmly bound to the end of the rod with a cotter-key and gib, which also controls the adjustment for wear.

Fig. 274 has strap ends front and back. Keys are inserted between the straps and the rod to prevent the shear of the strap-bolts. The construction of this rod and the method employed to take up the wear are plainly shown in the figure. The Erie City Iron Works use this rod on their stationary engines.

Exercise 132.—Make the drawings as shown in Fig. 273. (*Scale 6″ = 1 foot.*)

Exercise 133.—Make the drawings as shown in Fig. 274,

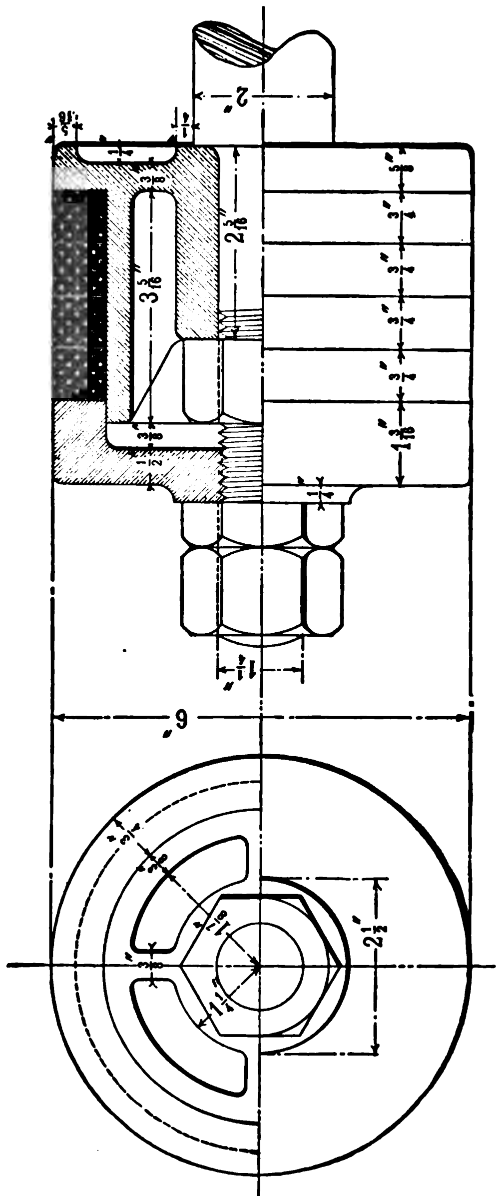


FIG. 272.

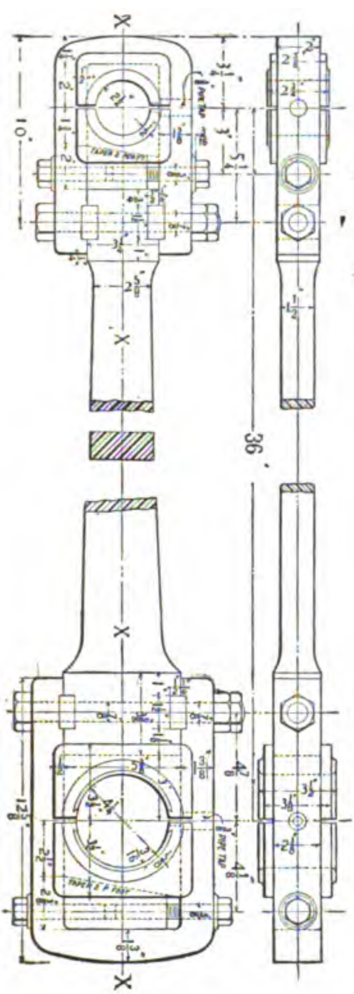


FIG. 274.

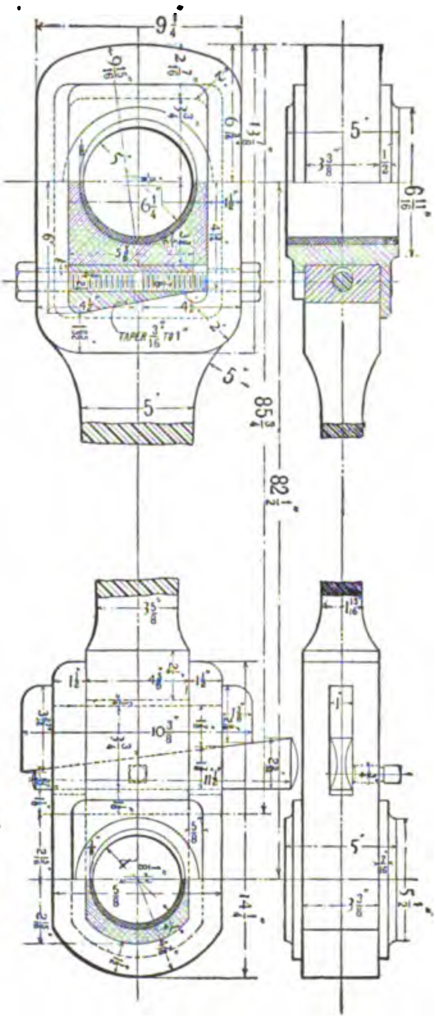


FIG. 273.

except that half of the plan shall be a section through *XX*. (*Scale 6″ = 1 foot.*)

Fig. 275 is the connecting-rod used by the Pennsylvania Railroad Company on their fast passenger-locomotives. The crank end of this rod is an improved design invented by Mr. A. S. Vogt, mechanical engineer of the company. He explains the improvements as follows:

As before, the back end of the rod is forked, but the method of closing the open end of the fork is entirely different, and the key for closing the main brasses has been moved from the forward side of the brass to the rear, which has another good effect, viz., as the brasses in both front end and back end of the rod wear and are closed up to meet that wear, the actual length of the rod changes but very little, for the reason that the keying of both ends is in the same direction, whereas in the old form of the rod the keying was in opposite directions, and as a matter of course the distance from centre of crank-pin to centre of crosshead-pin increased gradually. The open end of the fork in this rod is closed, first, by a U-shaped block, the detail of which is marked *A* on Fig. 275; next, by the key which is marked *B*; and last of all by a combined key and bolt marked *C*; this bolt clamping the two members of the fork against the block *A* and forming an enclosed surface for the key to drive against. To prevent the slacking up of the nut *C*, a keeper-block is provided at the bottom of the lower member of the fork. This is made with a recess into which the nut fits and a set-screw for locking the nut. The same keeper-block extends forward to the key *B*, which is also blocked by a set-screw in the block. It is quite evident that there is much less chance of shearing

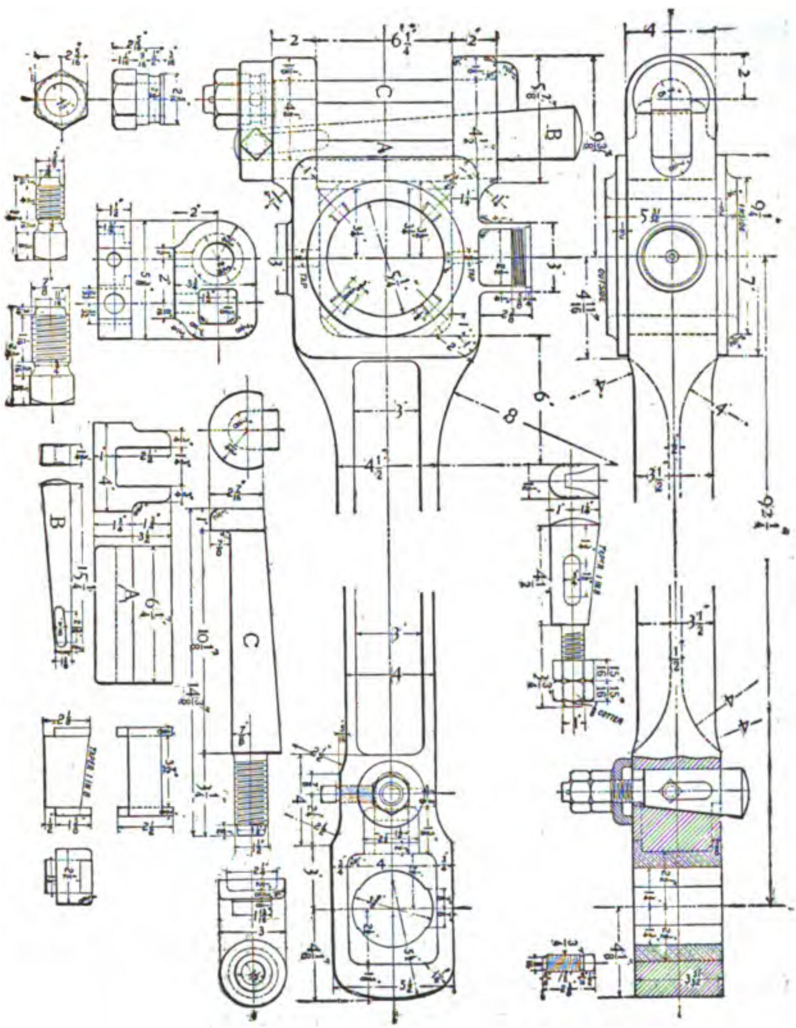


FIG. 275.

or offsetting of the bolt and the key in this than there was of the bolt in the former design; but even if it should take place, which is not very likely, the whole thing can readily be disconnected by, first, driving the key *B* out, unscrewing the nut on the bolt and moving the whole bolt a little bit forward, when it can be lifted out at the top.

Exercise 134.—Make drawings as shown in Fig. 275. (*Scale 6″ = 1 foot.*)

Fig. 276 is a marine connecting-rod, and is in use on the U. S. cruiser "Olympia."

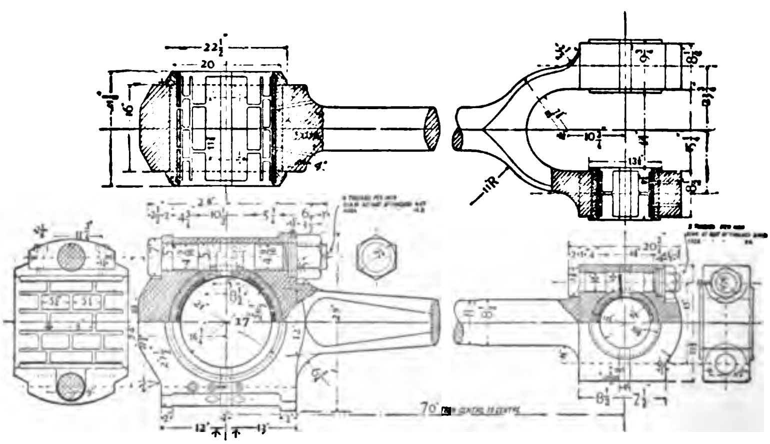


FIG. 276.

The general construction of this rod and the method used for taking up the wear are plainly shown in the figure.

A detail drawing of the bolt and its locking arrangement is given in Fig. 65, page 96.

Exercise 135.—Make drawings as shown in Fig. 276. (*Scale 2″ = 1 foot.*)

Thrust of Connecting-rod.—Assuming that a connecting-rod is equal to a pillar rounded or jointed at both ends,
let $D =$ diameter of piston in inches;
$L =$ length of stroke in inches;
$l =$ length of connecting-rod in inches;
$P =$ maximum steam-pressure per square inch;
$T =$ thrust of connecting-rod.

When the crank-pin is on a dead-centre and the connecting-rod is in line with the piston-rod, then

$$T = \frac{\pi}{4} D^2 P = W,$$

the total load on the piston. But as the crank rotates the connecting-rod becomes inclined to the centre line of motion, and T increases as the angle of the connecting-rod increases until a maximum is reached at half-stroke, provided the steam is not cut off before.

The value of T may be found for any position of the crank as follows:

Let AB, Fig. 277, be the connecting-rod, and BC the crank. The forces acting at A are W, the maximum pressure on the piston, and R, the reaction of the guide on the cross-head, and T, the thrust along the connecting-rod.

From the triangle of forces

$$\frac{T}{W} = \frac{AB}{AC}$$

and

$$T = W\frac{AB}{AC} = W\frac{AB}{\sqrt{AB^2 - AC^2}} = W\frac{l}{\sqrt{l^2 - \frac{L^2}{4}}} = W\frac{2l}{\sqrt{4l^2 - l^2}}.$$

Diameter of Connecting-rod, Circular Section.—Thurston gives

$$d = a\sqrt{Dl_1\sqrt{P}} + C = \text{diameter at middle,}$$

where $a = \begin{cases} 0.15 \text{ for fast engines,} \\ 0.08 \text{ for moderate speed;} \end{cases}$

$C = \begin{cases} \frac{1}{2}'' \text{ for fast engines,} \\ \frac{3}{4}'' \text{ for moderate speed;} \end{cases}$

$l_1 =$ length of connecting-rod in feet.

Seatons, Marks, and Whitham give

$$d = 0.02758\sqrt{Dl\sqrt{P}}.$$

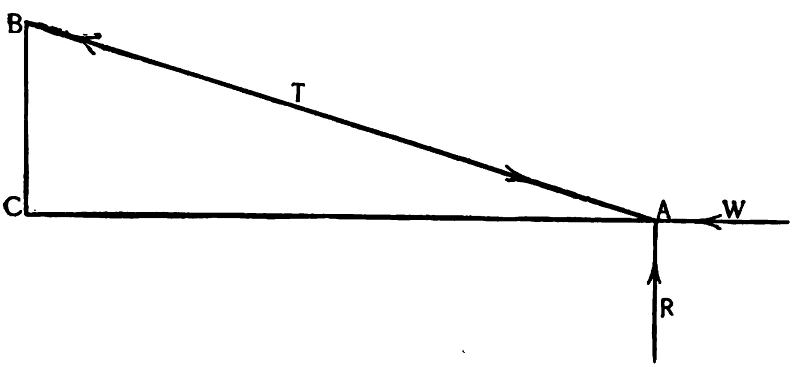


FIG. 277.

For the diameter at the crank-pin end Whitham gives 1.08 times the diameter at the cross head end. The rod is larger at the middle and tapers about $\frac{1}{8}''$ to the foot.

Sennett gives diameter at middle $= \dfrac{D}{55}\sqrt{P};$

" " necks $= \dfrac{D}{60}\sqrt{P}.$

Locomotive Connecting-rods.—The sizes of rectangular rods of uniformly tapered section are in practice as follows:

Depth of Main Rod.—On engines with cylinders 14″ diameter or less the depth of the rod at the crank end is made $\frac{5}{8}$″ less than the depth of the stub; over 14″ diameter, $\frac{7}{8}$″ less.

Depth of main rod at cross-head end $= d_1$.

Cyl. diam.	14″	15″	16″	17″	18″	19″	20″
d_1	$2\frac{3}{4}$″	$2\frac{3}{4}$″	3″	3″	3″	3″	$3\frac{1}{4}$″

Thickness of main rod $= t_1$.

Cyl. diam.	14″	15″	16″	17″	18″	19″	20″
t_1	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$1\frac{7}{8}$″	$1\frac{7}{8}$″	2″	2″	2″

Depth of Side-rods.—The depth of the side-rod is made about $\frac{5}{8}$″ narrower than that of the stub end, and of uniform depth throughout.

Thickness of side-rods $= t_2$.

Cyl. diam.	14″	15″	16″	17″	18″	19″	20″
t_2	$1\frac{5}{8}$″	$1\frac{5}{8}$″	$1\frac{5}{8}$″	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$1\frac{3}{4}$″

Pivot or Step Bearing.—In this form of bearing the pressure is applied in the direction of the axis, and the load is carried entirely upon the end of the shaft. In lubricating a bearing of this type the oil should always be introduced between the bearing surfaces from the under side and in the centre of the bearing, that the oil, under the influence of the

centrifugal force, will be distributed over the entire rubbing surface. The best results are obtained from running the bearing in a bath of oil, as shown in Fig. 278, where the oil-basin surrounds the journal-box, which can be kept submerged in oil, thus insuring constant and efficient lubrication. In Fig. 278 the oil passes from the oil-basin *OB* to the journal-box through the hole *h*, and on to the under side of the shaft through the hole in the disk *BD*. To insure a continuous flow of oil through the holes, grooves are made on the under side of the journal-box, and upon the upper side of the projection upon which the disk *BD* is carried. To prevent lateral motion the end of the shaft is turned to fit the journal-box *J*, which is provided with a brass bush (*B*). The bush is secured against turning with the shaft by the projections *E*, which fit between lugs cast on the inner surface of the journal-box. The under side of the journal-box *J* is slightly spherical and rests upon a surface which is also slightly spherical. This allows the journal-box to be tipped over for a limited distance by means of the set-screws *S* until its axis coincides with that of the shaft. The down-pressure of the shaft is carried upon a steel disk *BD*, which also rests upon a slightly spherical projection cast on the upper side of the journal-box bottom, allowing the whole of the shaft end to remain in contact with the disk although the bush *B* has become sufficiently worn to allow the shaft to have side motion, thus making the bearing adjustable, and capable of maintaining a perfect bearing over its entire surface under all ordinary working conditions. The disk *BD* is longer than the shaft diameter, and is prevented from turning with the shaft by the flat sides coming in contact with the lugs *l*. The journal-box is hexagonal

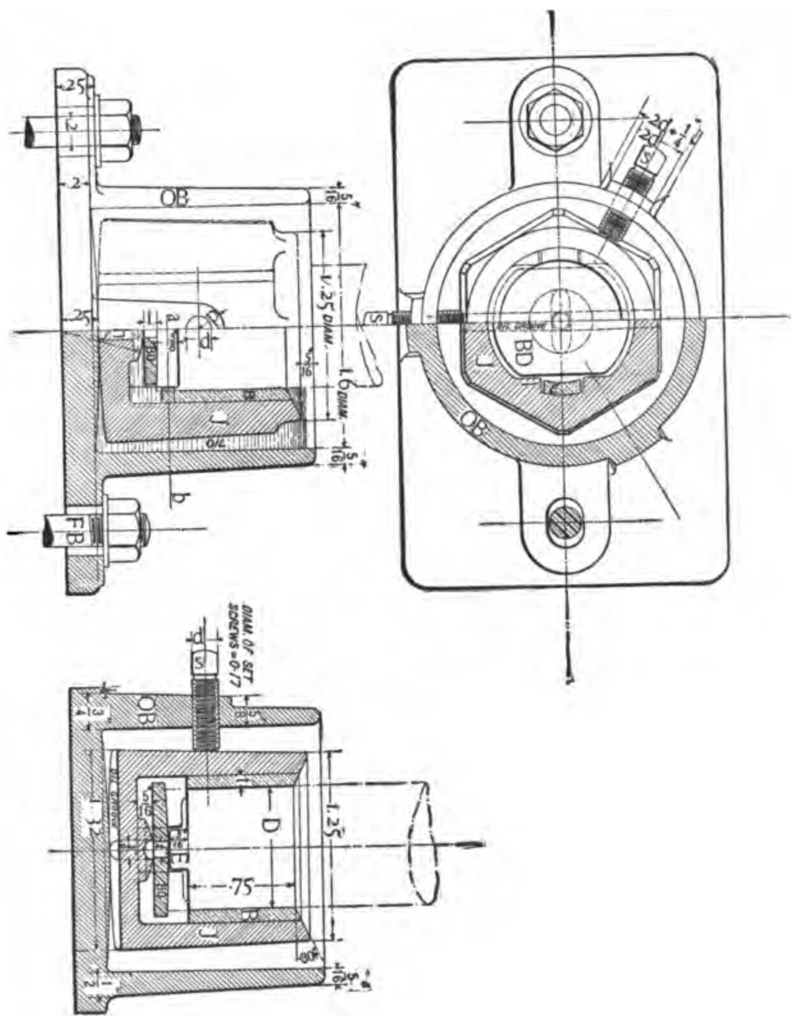


FIG. 278.

in cross-section, and is tapered towards the top to allow it to tip the required distance without increasing the diameter of the oil-basin. It is held against turning with the shaft by the set-screws S pressing against the flat faces. This form of pivot or step bearing is suitable for journals from $1\frac{1}{2}''$ to $3\frac{1}{2}''$ in diameter.

As the velocity of the bearing surface varies from zero at the centre to a maximum at the circumference, and as the friction increases with the velocity, the wear will increase from the centre to the circumference. Thus it will be seen that the smaller the diameter D of the journal, within limits determined by the pressure per square inch on the rubbing surfaces, the more will the tendency to wear be reduced.

D will be found by the formula

$$\frac{\pi D^2}{4} = \frac{T}{P},$$

from which

$$D = \sqrt{\frac{T}{.7854P}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where P = intensity of pressure per square inch of projected area, which with this form of bearing running continuously may be taken at 300 lbs.;

T = total load on the rubbing surface, which is the weight of the shaft and its attachments.

Exercise 136.—Design a bearing of the form shown in Fig. 278, to carry a load of 1450 lbs. Show a HALF ELEVATION, a HALF SECTIONAL ELEVATION, a SECTIONAL END VIEW, the planes of section passing through the centre of the bearing, a HALF PLAN and a HALF SECTIONAL PLAN, the plane of sec-

tion passing through the bearing at the line *ab*. The unit of proportions $= D + \frac{1}{2}''$. The thickness *t* of the brass bush, and the bearing disk at the centre may be made $= .08D + \frac{1}{16}''$. The parts dimensioned in inches are constant for all sizes of journals. *Scale full size.*

Crank-shaft or Main Bearings.—The bearings carrying the crank-shaft of a vertical engine have the greatest pressure acting nearly vertically; consequently the greatest wear will be above and below the shaft, and adjustment is effected by a two-part bearing, parted on the horizontal centre line, as in Fig. 170. The crank-shaft bearings of horizontal engines should be designed for horizontal adjustment to take up the side wear caused by the pull and thrust transmitted along the connecting-rod, and vertically to take up that caused by downward pressure due to the weight of the fly-wheels, etc. Vertical and horizontal adjustment can be obtained, with a two-part bearing, by parting the bushing at an inclination with the direction of both pressures. The inclination is generally made at 45°. The frame-work connected with the crank bearings on horizontal engines is generally part of the engine frame, as in Figs. 279 and 280.

Three-part Bearing.—An example of this form of bearing is shown in Fig. 279, where the horizontal wear is taken up in one direction only, by screwing in the screws *A*, which move up the adjusting-gibs *G* against the shaft. The vertical wear is taken up by screwing down the cap *C*. In this design, as the bearings wear, the shaft will be moved forward and down, and can only be returned to its original position by renewing the babbitt strips. The cap is made to fit into the frame, and is also provided with projections which fit over the outside of

the frame, thus insuring that it will sit squarely upon its journal. To keep the cap (C) from being screwed too far and

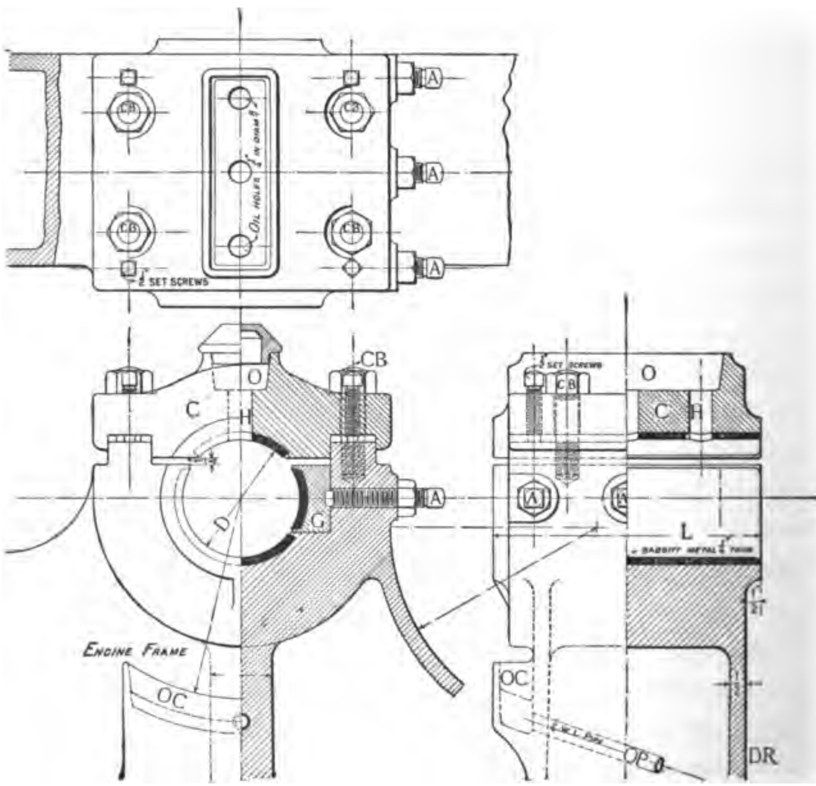


FIG. 279.

clamping the shaft, it is provided with an adjusting-screw at each corner.

The lubricator O consists of a pocket cast in the cap from which the oil is conveyed to the bearing through the holes H. These are filled with cotton to keep the oil from flowing into the bearing too rapidly. This system of lubrication is efficient,

but very wasteful unless the surplus oil flowing from the bearing can be caught and used again. This is done (Fig. 279) by casting a hollow projection *OC* on the frame under the bearing, from which the oil is drained off by the pipe *OP* to the bottom of the engine frame.

Four-part Bearing.—This form of bearing (Fig. 280) is parted on each quarter of the journal, which allows the wear, caused by the thrust and pull on the connecting-rod, to be taken up on either side. This is effected by screwing down the bolts *A*, which pull up the tapered wedges *W*, moving the gibs *C* forward toward the journal. To hold the bolts *A* against turning back, they are provided with a locking arrangement, shown, drawn to an enlarged scale, in Fig. 281. The vertical adjustment is obtained by screwing down the cap-bolts *CB*. The under side of the journal is carried upon a block *LB*, which is allowed to move transversely, thus allowing it to adjust itself to the journal, but is held against moving longitudinally by projections which fit over the raised part *F* on the frame. The top bearing *TB* is held in position by the screws *S*, which also serve to hold the gibs (*G*) in position and keep the cap from being screwed down too tightly on the shaft. The lubricator *O* may be used for semi-liquid grease or by filling it with cotton saturated with oil.

Length of Crank Bearing.—To calculate the length of a bearing it is necessary that we should know the amount and direction of the pressure to which it is subjected. The pressure on the crank-shaft bearings of a horizontal engine is uncertain in amount and direction. We can determine the amount and direction of the resultant pressure caused by the thrust and pull of the connecting-rod and that due to the

weight of the fly-wheels and shaft, but this pressure may be either augmented or relieved by the transmission of the power.

A reliable rule, and one which is generally observed in this country, is to make the length of the bearing equal to TWICE THE DIAMETER OF THE SHAFT.

The Cap.—To relieve the cap as much as possible from the stresses the frame is carried up well around the bearing, and the cap (C) is practically a flat plate. The upward pressure of the cap caused by the angularity of the connecting-rod is found by the formula

$$p = \frac{P}{\sqrt{R^2 - 1}}.$$

This pressure may be augmented by the gearing which is used to transmit the power, and, to insure that the cap and cap-studs will have sufficient strength under the worst conditions, the value of p should be increased 100%. Then the maximum pressure p' on the cap will be found by the formala

$$p' = \left(\frac{P}{\sqrt{R^2 - 1}}\right)2, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $P =$ total steam-pressure on the piston;

$R =$ ratio of length of connecting-rod to throw of crank.

The length of connecting-rod is generally made equal to 6 times the throw of crank. The cap is in the condition of a beam on which the load is distributed over its entire surface.

Then the bending moment is $\frac{p'l}{8}$, and the moment of resistance to bending is $\frac{LT^2}{6}f$. Therefore

$$\frac{p'l}{8} = \frac{LT^2}{6}f,$$

from which

$$T = \sqrt{\frac{p' \times l \times 6}{L \times 8 \times f}}, \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where L = length of cap;
T = thickness of cap;
p' = total load on cap;
l = distance between cap-studs;
f = strength of the material, which may be taken at 5000 lbs.

Diameter of Studs.—The maximum pressure (p') on the under side of the cap is resisted by the studs CB. Therefore their effective area will be found by the formula

$$\text{Area at bottom of thread} = \frac{p'}{n \times f_t},$$

where n = number of studs;
f_t = strength of material = 3000 lbs. per square inch of area at bottom of threads.

Having found the area at the bottom of the threads, turn to Table No. 8, page 66, from which take the nearest diameter of screw having the required area. The diameter of the adjusting-studs (A) and the set-screws (s) may be made $\frac{5}{8}''$ in diameter when the journal is 6″ or less, and increased $\frac{1}{8}''$ for every inch the journal is increased above 6″ in diameter.

The Gibs.—The height of the gibs (G) should be $\frac{3}{8}$, and their thickness at t should be equal to $\frac{1}{8}$, of the shaft diameter.

Adjusting-wedges.—Instead of using three adjusting wedges and screws, as in Fig. 280, another arrangement is to

use one wedge and one adjusting-screw with two guide-pins, as in Fig. 282. In the latter arrangement the wedge supports the gib and is in contact with the frame its entire length. The thickness of the wedges at the top should be $1\frac{1}{2}$ times the diameter of the screw (A) + $\frac{1}{8}''$, and their width w when

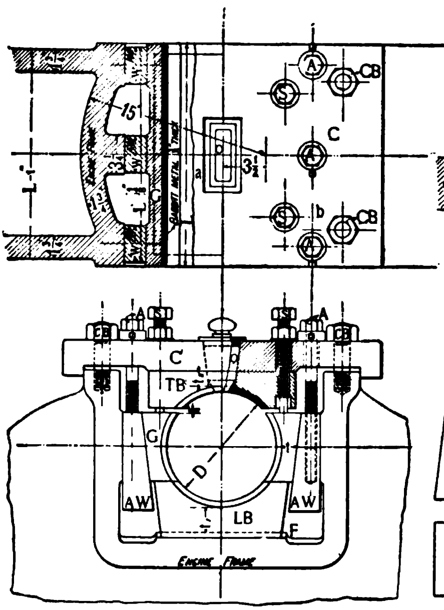


FIG. 280.

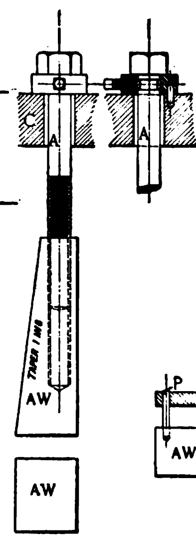


FIG. 281.

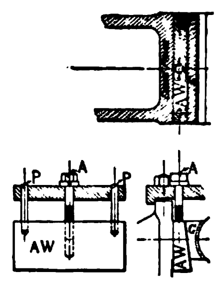


FIG. 282.

three are used should not be less than $\frac{1}{6}$ the length of the journal (L). The taper of the wedges may be made from 1 in 6 to 1 in 8. The screw A should be sufficiently long to enter the wedge W a distance equal to its diameter when the wedge is full down.

Top and Bottom Blocks.—The thickness t at the thinnest part of the bottom block should be equal to .23, and that of the top block .15, of the journal diameter.

Exercise 137.—Design a crank-shaft bearing of the form shown in Fig. 279, proportioned for a horizontal steam-engine, having a cylinder 9″ in diameter, stroke 10″, initial steam-pressure 200 lbs. per square inch, and the diameter of the journal (D) 4″. The bearing to have a vertical and horizontal adjustment of ⅜″. Show a HALF ELEVATION, A HALF SECTIONAL ELEVATION, a HALF END VIEW, a HALF SECTIONAL END VIEW, a HALF PLAN, and a HALF SECTIONAL PLAN of the right-hand side. *Scale 8″ to the foot.*

Exercise 138.—Design a crank-shaft bearing of the form shown in Fig. 280, proportioned for a horizontal steam-engine having a cylinder 18″ in diameter, stroke 30″ long, and an initial steam-pressure of 220 lbs. per square inch. The bearing to have a horizontal adjustment of ¼″ in either direction and a vertical adjustment of ⅜″. Make D the diameter of the journal 9″.

Show an ELEVATION, PART PLAN, and PART SECTIONAL PLAN, the plane of section passing through the centre of journal. *Scale 4″ to the foot.*

Show also a detail drawing of the adjusting screws and wedges, as in Fig. 281. *Scale 8″ to the foot.*

Exercise 139.—Design a crank-shaft bearing of the form shown in Fig. 280, substituting the adjusting-wedge arrangement shown in Fig. 282. Make the proportions suitable for the conditions given in Exercise 138. *Scale 4″ to the foot.*

Ball Bearings.—This device for reducing friction consists of perfect spheres placed between the journal and the bearing; the balls taking the place of the bush in supporting the shaft, thus substituting rolling for sliding friction. As the bearing areas are only slightly flattened points, the wear will

be comparatively rapid; so, to reduce the amount to a minimum, the balls and the surfaces upon which they roll are made of steel tempered as hard as possible.

The different forms of ball bearings are designated according to the number of points that the balls have in contact with the surfaces upon which they roll.

In a three-point bearing a line drawn through one of the points in the direction in which the load acts should pass midway between the other two points. Thus the form of bearing shown in Fig. 283 will give good results only when the resultant of all the pressures acts at an angle of 45°, otherwise the balls will not revolve on a true axis, but will have a screw motion and therefore a considerable amount of friction. The design shown in Fig. 284 is suitable for a pressure in a vertical direction only. In a four-point bearing a line drawn through one of the points in the direction in which the pressure acts should pass through a contact-point on the other side of the ball (as in Fig. 285), then the balls revolve on a true axis and sliding friction is entirely avoided.

Size of Balls.—Steel balls rolling under pressure do not fail by crushing, their period of usefulness depending upon both speed and pressure. This would seem to indicate that the balls should be as large as possible, thus reducing the number of revolutions in proportion to those of the shaft, and increasing their strength; but there is a practical limit to this owing to the fact that the larger the balls the fewer will be the number, and therefore the fewer the number of bearing points. The bearing would then fail by the balls crushing into the surfaces upon which they roll. There is a great diversity of opinion as to the proper size of ball in relation to

load and speed. The size given in Table No. 37 gives a fair average proportion of the diameter of the ball to the diameter of shaft used in practice for horizontal bearings.

TABLE NO. 87.

Shaft Diam.	Ball Diam.	Crushing Strength of Ball.	Shaft Diam.	Ball Diam.	Crushing Strength of Ball.
$\frac{1}{4}''$	$\frac{3}{16}''$	3 000	$1\frac{1}{4}''$	$\frac{1}{2}''$	20 000
$\frac{3}{8}$	$\frac{3}{16}$		$1\frac{1}{2}$	$\frac{1}{2}$	
$\frac{1}{2}$	$\frac{1}{4}$	5 000	$1\frac{3}{4}$	$\frac{5}{8}$	30 000
$\frac{5}{8}$	$\frac{1}{4}$		2	$\frac{5}{8}$	
$\frac{3}{4}$	$\frac{5}{16}$	7 000	$2\frac{1}{2}$	$\frac{3}{4}$	40 800
$\frac{7}{8}$	$\frac{5}{16}$		3	$\frac{3}{4}$	
1	$\frac{3}{8}$	12 000	$3\frac{1}{2}$	$\frac{7}{8}$	50 000
$1\frac{1}{8}$	$\frac{3}{8}$		4	1	60 000

Ball Races.—The thickness (t) of the surfaces upon which the balls roll should not be less than $\frac{1}{2}$ the diameter of the ball, and the width $W = 1\frac{1}{2}$ times the ball diameter. The angles of the grooves are generally made 45°. In every ball race a slight amount of clearance (c) is left between each pair of balls. This is necessary, first to get the balls into place, second to insure the free rolling of the balls. The amount of clearance is generally made from .002 to .004; it will therefore be safe to assume .003 as good practice. Then taking the diameter D_1 of the ball circle $= D + 2t = D + d$. In Fig. 285 the angle $\theta = \frac{360°}{2n} = \frac{180}{n}$, $\sin \frac{180°}{n} = \frac{d + c}{D_1} = x$.

From a table of sines find the angle θ in degrees corresponding to x. Then

$$\frac{180°}{n} = \theta \quad \text{and} \quad n = \frac{180°}{\theta}. \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

The number of balls must be within $.001 \times n$ of being a whole number; if not, we must increase the diameter D_1. Thus

supposing that formula No 4 gives $n = 20.75$, then we must increase D_1 to get in the next whole number of balls.

Taking $n = 21$, we can find D_1 by the formula

$$D = \frac{d + c}{\sin \frac{180^\circ}{n}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Load on Bearings.—As already explained, the life of a bearing is a function of both speed and load. Therefore if

FIG. 283.

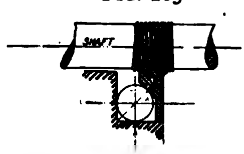


FIG. 284.

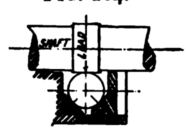
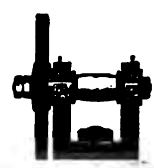


FIG. 286.



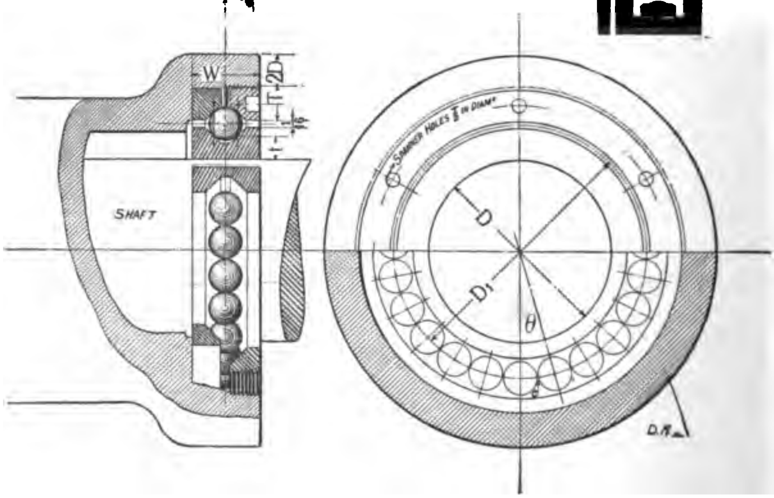


FIG. 285.

the speed is increased, the load must be correspondingly decreased or the life of the bearing will be shortened. Using the proportion of ball to the shaft diameter given in Table

37, the safe load in relation to the speed may be found by the formula

$$L = \frac{f_c \times N}{S}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

where L = total load on the bearing;
f_c = strength of ball;
S = speed of the ball races in feet per minute;
N = number of balls carrying the load; in horizontal bearings = $\frac{1}{4}$ of the total number.

Exercise 140.—Design a lathe grinder of the form shown in Fig. 286 provided with four-point ball bearings. Make the emery-wheel 6″ in diameter $\times \frac{1}{2}$″ thick, belt drum $1\frac{1}{4}$″ in diameter and length suitable for a $1\frac{1}{8}$″ belt. The diameter of the shaft $(D) = \frac{5}{8}$″.

Show an ELEVATION with one of the bearings partly in section, a HALF END VIEW and a HALF SECTIONAL END VIEW as shown in Fig. 286. *Scale twice full size.*

Thrust Bearings.—The difficulty experienced with the ordinary pivot or thrust bearing, due to the velocity increasing as the distance from the centre, is overcome by the application of balls to this type of bearing. The designs shown in Figs. 287 and 288 are made by the Boston Ball Bearing Co. and may be used on either vertical or horizontal shafts. The balls are held in place by the cage C, the use of which, although tending to increase rather than diminish friction, facilitates the placing and removing of the balls, and by its use the balls can be placed at various distances from the centre of the shaft, thus increasing the time the bearing will run before wearing grooves in the plates PB. In Fig. 288

the balls are arranged in spirals; thus every ball runs on a separate path, and the tendency to wear grooves is reduced to a minimum. The small cages are made in one piece, as in Fig. 287, and the balls are put into position by springing the cage, while in the large cages the top is fastened to the under side, by rivets, after the balls are in position, as in Fig. 288. The cages are made from $\frac{3}{32}''$ to $\frac{1}{8}''$ thick. The thickness (t) should not be less than $\frac{1}{3}$ the diameter of the ball, and the

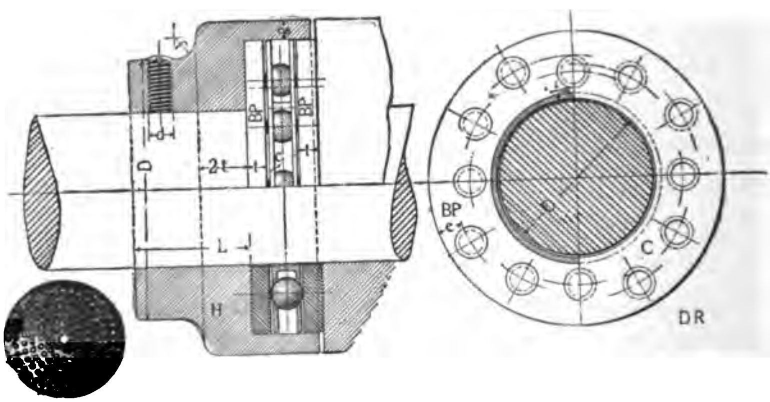


FIG. 288. FIG. 287.

distance e should not be less than $\frac{3}{4}$ of the ball diameter. The centres of the ball races are $\frac{1}{4}$ of a ball diameter apart. The hub H is screwed to the shaft by means of one or more set-screws (S), the diameter of which may be made $.2D$, but not greater than $\frac{3}{4}''$. $L = 2t + 3d$, but not less than $\frac{3}{4}D$. $g = .12D$.

$D' = 2D$ when D is less than $2''$, and $= 1.7$ when D is $2''$ or over. The load and speed to which this type of bearing is subjected will determine the number of balls. Taking the size

of ball in proportion to the diameter of the shaft from Table No. 37, then from formula No. 6

$$N = \frac{L \times S}{f_c}.$$

Exercise 141.—Design a thrust bearing of the form shown in Fig. 287 for a 4″ shaft, to carry a load of 760 lbs. and run at a speed of 600 revolutions per minute. *Scale full size.*

Stuffing-boxes.—To prevent leakage, when rods work through the walls of a chamber containing fluid, the rod is passed through a cavity filled with an elastic material which will adjust itself to any irregularities on the surface of the rod. Fig. 289 shows a stuffing-box suitable for a horizontal

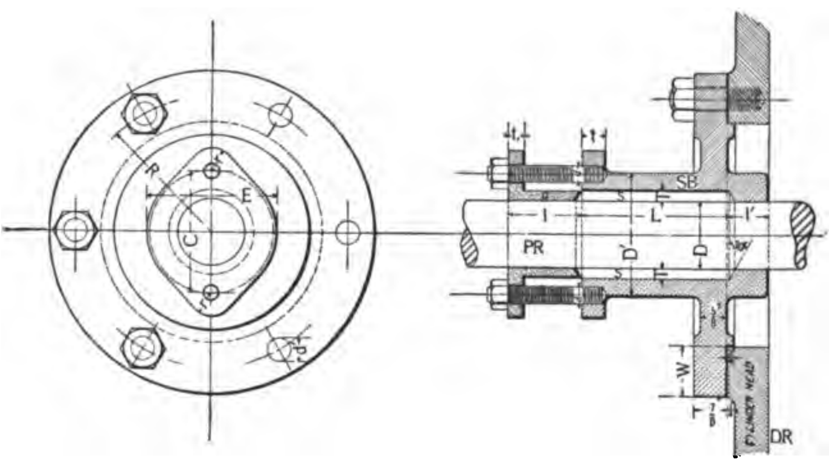


Fig. 289.

steam-engine piston-rod, and Fig. 290 one arranged for a vertical steam-engine piston-rod.

The stuffing-box *SB* may be made a separate piece and bolted to the cylinder-head, as in Fig. 289, or cast with the cylinder-head, as in Fig. 290. Part of the box *SB* is bored

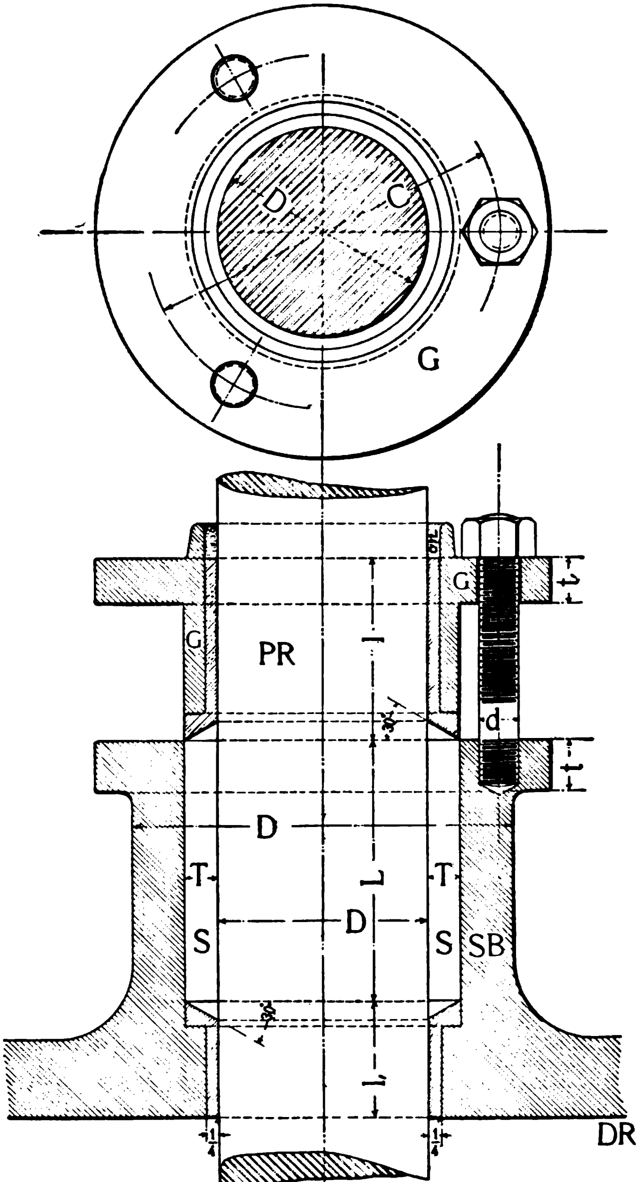


FIG. 290.

larger than the diameter of the piston-rod PR, thus leaving a space S around the rod which is filled with packing consisting of a fibrous material saturated with oil or tallow. The packing is pressed against the rod by screwing down the gland G, which is generally made of brass for rods under 4" in diameter, as in Fig. 289, and of cast iron lined with brass for the larger rods, as in Fig. 290.

Proportions.—The proportions of the stuffing-box are generally decided by the conditions under which it is used; thus the box is generally made longer for a high than a low pressure. However, under any conditions, the longer the box the longer will the packing last.

The following proportions are suitable for average pressures and speeds, and could be used for high pressure, but would require to be repacked comparatively often:

$L = 2D$ for rods 2" or less in diameter;
$L = 1\frac{1}{2}D$ for rods between 2" and 3" in diameter;
$L = 1\frac{1}{4}D$ " " " 3" " 4" " " ;
$L = D + 1''$ for rods over 4" in diameter;
$l = .75L$; $T = \frac{1}{8}D$ in nearest $\frac{1}{16}$;

$D' = 1.75D + .25$;	$l = .5D + \frac{1}{4}''$;
$d = .2D$;	$r = d + \frac{1}{4}''$;
$C = 1\frac{1}{8}D + 2d$;	$E = D' + \frac{1}{8}''$;
$t = d + \frac{1}{4}''$;	$t_1 = d$;
$R = 2D$;	$d' =$ from $\frac{5}{8}''$ to $\frac{3}{4}''$.

Exercise 142.—Draw a stuffing-box, in which soft packing is to be used, for a horizontal-engine piston-rod (Fig. 289). Make $D = 1\frac{5}{8}''$ in diameter. *Scale full size.*

Exercise 143.—Draw a stuffing-box (Fig. 290), in which

soft packing is to be used, for the H.-P. cylinder of a vertical steam-engine. Make $D = 4''$. Thickness of cylinder-cover $1\frac{1}{2}''$. *Scale 8″ to the foot.*

Metallic Packing.—Many designs of metallic packings have been devised to replace the soft packings. One of the most successful is that known as the United States Metallic Packing. A design showing the application of this form of packing suitable for high-pressure steam-engine piston-rods is shown in Fig. 291. This form is known as the "double packing" and is practically two sets of the ordinary form of packing arranged in tandem. In Fig. 291 the back packing is shown in section and the front partly in section. The packing consists of babbitt-metal rings *A*, *B*, and *C* which are cut in halves and forced into the cup *H* by the spiral spring *S*. On the packing nearer the cylinder the spring *S* will be aided by the steam-pressure acting on the follower *F*. The rings *A*, *B*, and *C* are conical, and being forced into the correspondingly shaped cup *H*, the cup-rings close and press against the piston-rod *PR*. The cup *H* rests against the flat face of the ring *R*, which forms a ball-and-socket joint with the outer casing *G* or preventer *P′*. As the cup *H* is free to slide on the flat face of the ring *R*, which in turn is free to rock on the casing *G* or *P′*, the packing never binds the rod nor constrains it in any way. The packing is prevented from drawing back with the rod (beyond a small movement) by the flange on the follower *F* coming in contact with the preventer *P* or *P′*.

Exercise 144.—Draw the arrangement of United States Metallic Packing shown in Fig. 291. *Scale full size.*

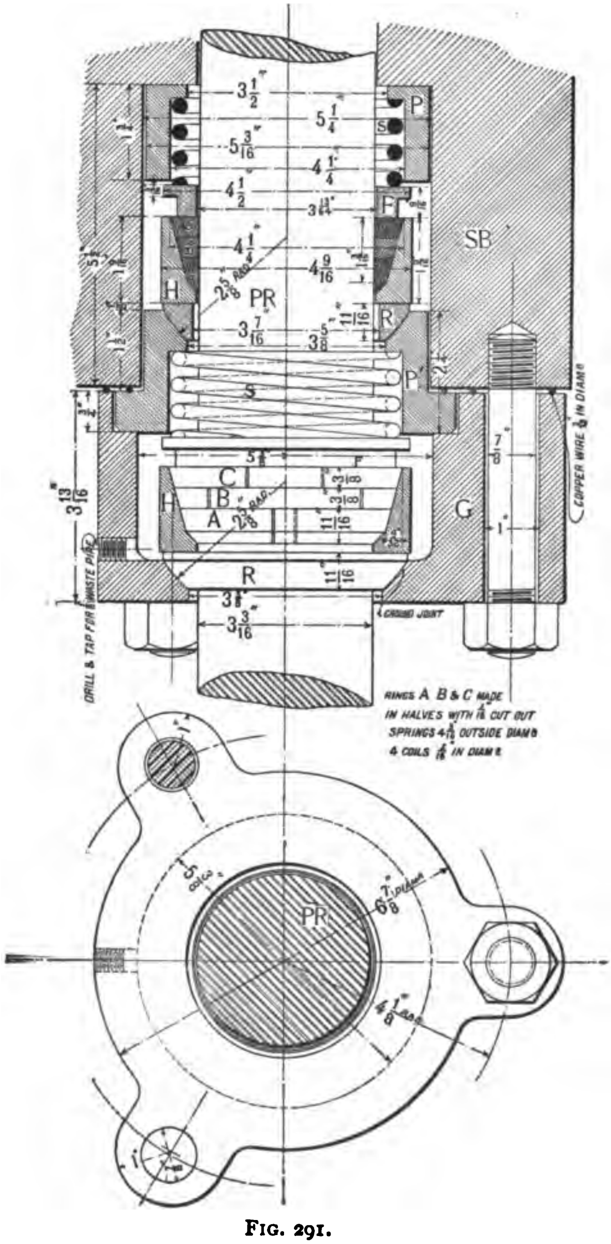

RINGS A B & C MADE IN HALVES WITH 1/8" CUT OUT
SPRINGS 4 1/8 OUTSIDE DIAM.
4 COILS 1/8" IN DIAM.
DRILL & TAP FOR WASTE PIPE
COPPER WIRE 1/8" IN DIAM.
GROUND JOINT
SB
PR
G

FIG. 291.

Cross-heads and Guides.—When the connecting-rod is inclined toward the direction in which the piston is moving it will exert an upward or downward pressure according to the direction in which the engine is running, and, unless special means are employed, would tend to bend the piston-rod or force it out of its straight path. To prevent such an occurrence the piston-rod end is provided with a cross-head which

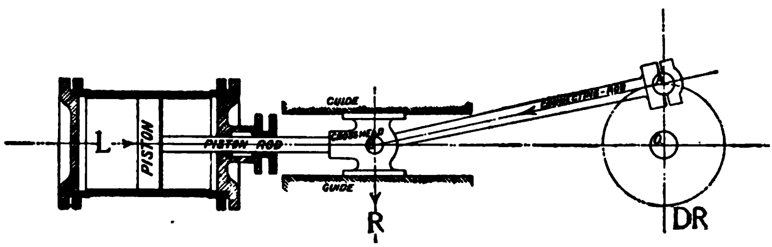


FIG. 292.

slides on surfaces that are parallel with the piston-rod, called guides.

Cross-head Blocks.—Assuming that steam is not cut off before midstroke, then the thrust caused by the obliquity of the connecting-rod will reach a maximum when the crank is nearly at right angles with the line BO (Fig. 292).

Taking $L =$ load on piston;
$R =$ thrust of the connecting-rod;
$p =$ steam-pressure per square inch;
$p' =$ intensity of pressure per square inch;
$V =$ velocity of cross-head in feet per minute;
$A =$ area of the bearing-surface in square inches,—

then

$$L : R :: BO : AO.$$

Therefore

$$R = L \times \frac{AO}{BO} = L \times \frac{AO}{\sqrt{BO^2 - AO^2}}.$$

Taking the length of the crank OA as the unit, then

$$R = L \times \frac{1}{\sqrt{n^2 - 1^2}} = \frac{L}{\sqrt{n^2 - 1}} = \frac{\frac{\pi d^2}{4} \times p}{\sqrt{n^2 - 1}},$$

where n = ratio of connecting-rod to crank.

Pressure on Rubbing-surfaces.—There is great diversity of opinion as to the proper intensity of pressure on the guide-blocks. It varies, according to the different authorities, from 22 to 500 lbs. per square inch. Thurston gives $p' = \frac{40000}{V}$, and that this value in marine and stationary engines may be exceeded to the extent that $p' = 60000 \div V$. Then

$$A = \frac{R \times V}{40000} = \frac{\frac{\pi d^2}{4} \times p \times V}{40000\,(\sqrt{n^2 - 1})}. \quad . \quad . \quad . \quad (7)$$

In many cases of ordinary stationary-engine practice, especially on engines having four-bar guides, the above formula would give a very short block, and as there is generally no difficulty in providing large rubbing-surfaces, we find the areas increased as large as

$$A = \frac{R \times V}{25000}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where V = velocity of piston in feet per minute = (length of stroke $\times$ twice the number of revolutions per minute).

The cross-head should always be designed so that the

resultant pressure (R) on the guides will have its point of resistance at the centre of the cross-head rubbing-surfaces, as shown in Fig. 292.

Wrist-pin.—The connecting-rod is attached to the cross-head by the pin CP, Fig. 293. In this form of joint, as the velocity is low and the pressure constantly changing in direction and magnitude, the allowable pressure per square inch is comparatively high, reaching in some designs as much as 1400 lbs. per square inch. Seaton says that the pressure per square inch should never exceed 1200 lbs. per square inch of projected area ($d \times l$).

When the total load on the pin is taken as the maximum load on the piston, i.e., the initial steam-pressure × area of piston, the length of the pin is generally made to equal from d to $1.3d$.

Taking the length $l = d$, then

$$d = \frac{\sqrt{L}}{p'}.$$

When the length of the pin is made equal to $\frac{4}{3}$ of its diameter, then

$$d = \frac{\sqrt{L}}{\frac{4}{3}p'}.$$

Taking the value of $p' = 1200$, then

$$d = \sqrt{L} \div 40 \quad \text{and} \quad l = \sqrt{L} \div 30. \quad . \quad . \quad (10)$$

A pin proportioned to either of the above formulæ will be amply strong to resist bending.

Guide-bars.—When the guiding-surfaces are part of the frame the guides are bored and the bearing-surfaces on the cross-head are turned as in Fig. 279. This arrangement

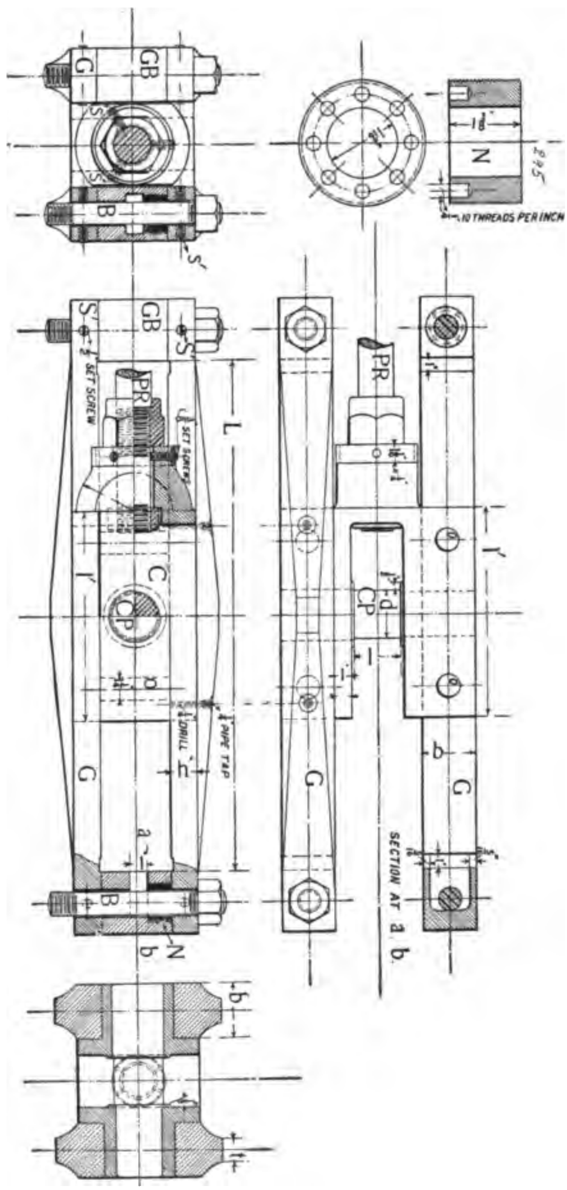


FIG. 293.

FIG. 295.

reduces the number of parts, which is always a good point in designing, as it not only decreases the labor but also the liability to error in fitting up. When made separate the bearing-surfaces are flat, and the guide-bars are generally of rectangular (when of steel) or T section (when of cast iron).

To prevent the formation of ridges, due to the travel of the cross-head varying as the wear on the connecting-rod joints is taken up, grooves are cut across the bars over which the ends of the cross-head blocks (CB) project at the end of each stroke.

Strength of Guides.—The greatest pressure on the bars occurs when the cross-head is nearly at the centre. Then the bending moment is $= \frac{RL}{6}$ and the moment of resistance to bending $= fZ$. Where Z is the modulus of section, given in Table No. 29.

The Length of Guide-bars between distance-pieces is = to the stroke + the length of the block + end clearance, which may be made = 1″ at each end.

Four-bar Guide.—The arrangement shown in Fig. 293 is that used on the cycloidal engine (Atlas Engine Works).

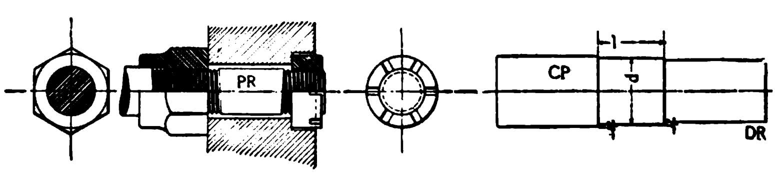


FIG. 294.

With this arrangement the pressure P is equally distributed on each side of the piston-rod, which is guided laterally as well as vertically by the cross-head sliding on the inner surfaces of

the guide-bars G. The piston-rod PR is secured to the cross-head C by the arrangement shown in Fig. 294. To prevent the piston-rod from exerting undue pressure on the stuffing-boxes, should the axis of the piston-rod not coincide with that of the cross-head, the hole through the shank of the cross-head is made larger than the diameter of the piston-rod, which is adjustable and held in position by means of the set-screws S. In this arrangement the breadth b of the bearing-surfaces on each bar is generally made equal to $\frac{1}{4}$ their length.

The area of the bearing-surfaces may be determined by formula No. 8. Then

$$b = \sqrt{\frac{.5A}{4}} \quad \text{and} \quad l' = \frac{.5A}{b}.$$

The form of guide-bar used in this design may be made of cast iron or steel, and proportioned in the following manner:

Having determined the breadth b and the length L, then calculating for a bar of rectangular section, secured at both ends and loaded at the centre, the height h of the bar at the centre will be found by the equation

$$\frac{RL}{6} = f\frac{bh^2}{6},$$

from which

$$h = \sqrt{\frac{R \times L \times 6}{b \times f \times 6}},$$

where f may be taken at 3000 for cast iron, and 6000 for steel.

Take $h' = .75\ h$, then the area of the web will be $=$ $(h \times b) - (h' \times b)$, and taking the thickness t of the web $= .4b$. Then the height of the web at the centre will equal area of web $\div\ 4b$.

The greatest strain on the stud-bolts *B* which secure the guides to the engine-frame is due to screwing up. They may be made $= 1\frac{1}{4}''$ in diameter. To allow for any slight inaccuracy of workmanship, the holes through the bars are made $\frac{1}{16}''$ larger than the diameter of the bolts *B*, and the bars are adjusted laterally by the screw *S'*. The bars are adjusted vertically by means of the nut *N*, shown in Fig. 295, which is screwed into the guide-bar blocks *GB*. The rubbing-surfaces are lubricated by oil-cups screwed on to the upper guide-bars. The oil is transmitted to the lower bars through the holes *O* on the cross-head.

Exercise 145.—Draw the four-bar guide and cross-head arrangement shown in Fig. 293 suitable for an engine having a cylinder 12″ in diameter × 15″ stroke. Initial steam-pressure 75 lbs. per square inch. Speed 300 revolutions per minute, and a connecting-rod four times the length of the crank. *Scale 4″ to the foot.*

Draw also details of the adjusting-nut *N* and the cross-head pin, and show the arrangement of fastening the piston-rod to the cross-head, taking the diameter of the piston-rod = 2″. *Scale full size.*

Two-bar Guide.—When two guide-bars are used they are arranged either one above and one below the piston-rod (in this case a cross-head of the type shown in Fig. 297 is used) or both guide-bars above the piston as shown in Fig. 296. The latter arrangement is one commonly used in locomotive construction.

The pressure is on the upper guide, *UG*, when the locomotive is running forward, and on the lower guide, *LG*, when running back; and as the engine is generally run forward more

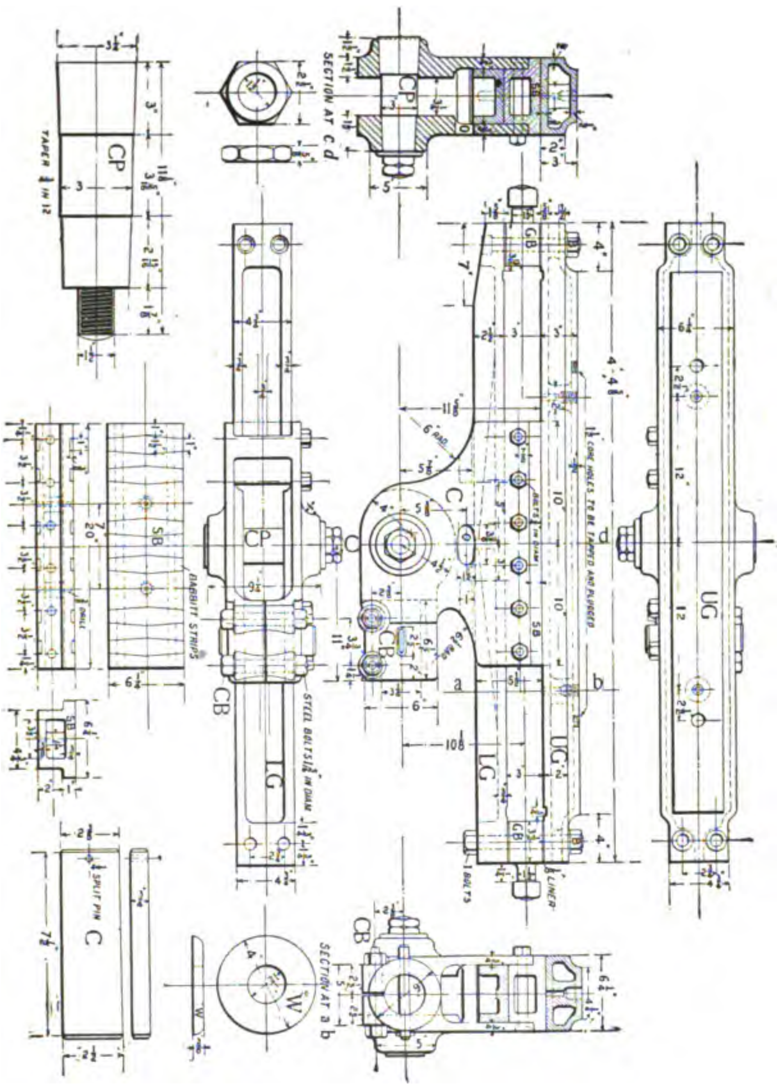

SECTION AT c d
SECTION AT a b
TAPER 1/4 IN 12
BABBIT STRIPS
STEEL BOLTS 1 1/4 IN DIAM
SPLIT PIN
CP
CB
LG
UG
GB
SB
C
W

FIG. 296.

than back, the bearing-surface on the lower bar may be made smaller than that of the upper.

In this design the cross-head *C* is of cast steel and provided with a brass slide-block *SB* which has strips of babbitt metal top and bottom. To fit and remove the piston-rod easily from the cross-head, the shank is cut and, after the rod is in position, it is gripped by screwing down the bolts *CB* and secured by driving a tapered cotter through it and the cross-head shank.

The hole *O* is to allow for lubricating the cross-head pin.

The guide-blocks *GB* are fastened to the cylinder at one end and to a guide-bar frame at the other.

Exercise 146.—Draw the cross-head and guide-bar arrangement shown in Fig. 296. *Scale 3″ to the foot.*

Also details of the slide-blocks *SB*, guide-block *GB*, cross-head pin *CP*, cotter *C*, and washer *W*. *Scale half size.*

Cross-heads.—Adjustments to take up the wear or for original setting may be accomplished by moving the guide-bars, as in Figs. 293 and 296, or the slide-blocks, as in Fig. 297.

In this design the cross-head *C* is hollowed to receive the connecting-rod end, which works upon the pin *CP*. The pin is of case-hardened steel and is kept from turning by the $\frac{1}{4}$-inch square-headed screw *K*.

The piston-rod *PR* is screwed into the cross-head and secured by the nut *LN*. The socket into which the rod *PR* is screwed has flat surfaces on the top and bottom to give clearance for the nut *N*.

The diameter at the end of the socket is equal to the dis-

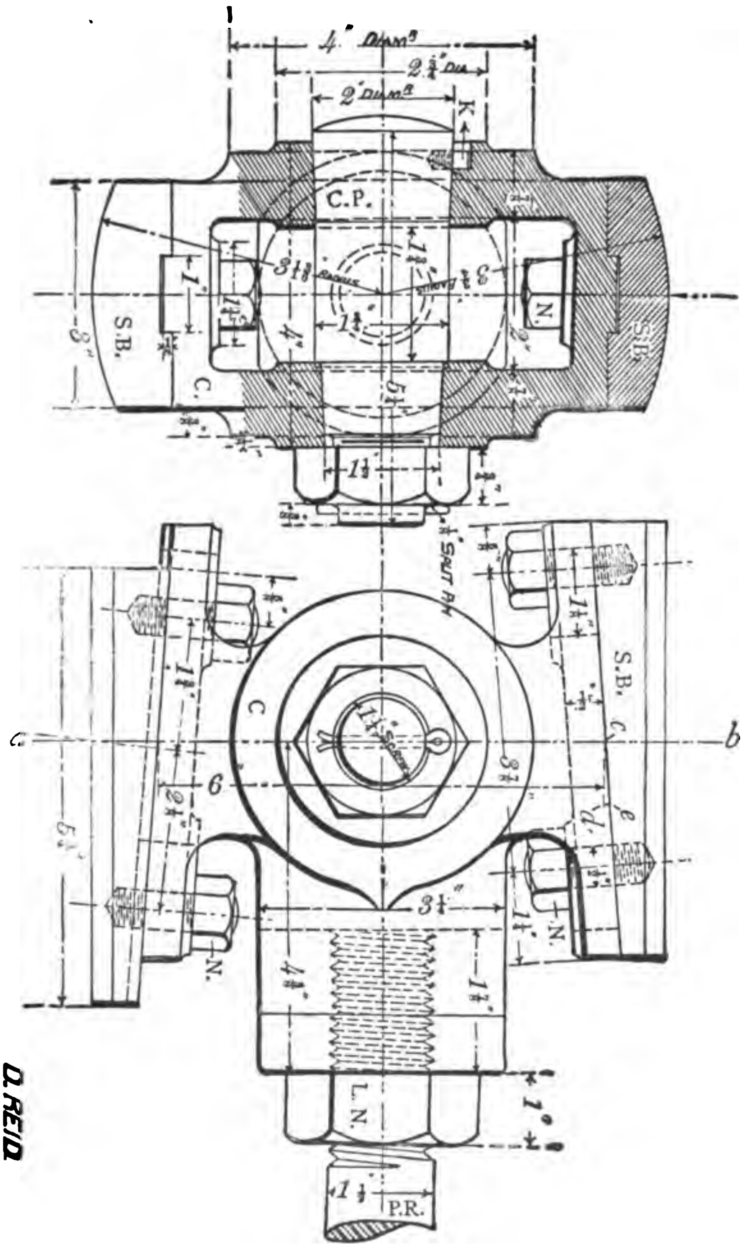


Fig. 297.

tance across the flats, and tapers back $\frac{3}{4}$ of an inch to the larger diameter.

The bearing-surfaces on the slide-blocks are turned, and the corresponding surfaces on the frame, upon which they fit, are bored to the same radius. The blocks are provided with grooves on the under sides, which fit over projections on the top of the cross-head, to prevent their lateral movement. To take up the wear, the slide-blocks move horizontally, on the inclined surfaces upon the top and bottom of the cross-head, for a distance equal to the length of the holes minus the diameter of the studs, and by this horizontal motion they move vertically $\frac{1}{16}$ of an inch.

Exercise 147.—Draw a cross-head of the form shown in Fig. 297, showing a SIDE ELEVATION, END ELEVATION PARTLY IN SECTION, and a SECTIONAL PLAN, the plane of section passing through the centre of the cross-head pin. *Scale full size.*

Construction.—To find the inclination necessary to give the required vertical movement, mark off on the centre line *ab* the distance from the centre of the pin *CP* to the point *C*, and through *C* draw the line *cd* at right angles to *ab* and equal to the horizontal motion of the slide-blocks, and through *d* draw *de* equal to the vertical movement.

The line drawn through the points *ce* will have the required inclination.

Fig. 298 shows a form of cross-head used on the U. S. cruiser Olympia. In this design the wrist-pin *CP* is outside of the cross-head, and there are two bearing-surfaces on the connecting-rod end. The slide-blocks *SB* are secured to the cross-head *C* by the bolts *B*. To allow the removal of the

slide-blocks while the cross-head is in position, one of the projecting lips L on each block is removable and held in place by the bolts B. To facilitate the removal of the piece L, it is provided with set-screws S. The piston-rod PR is secured to the cross-head by the nut shown in Fig. 67, page 100.

Fig. 299 is an isometric sketch of the complete cross-head.

FIG. 299.

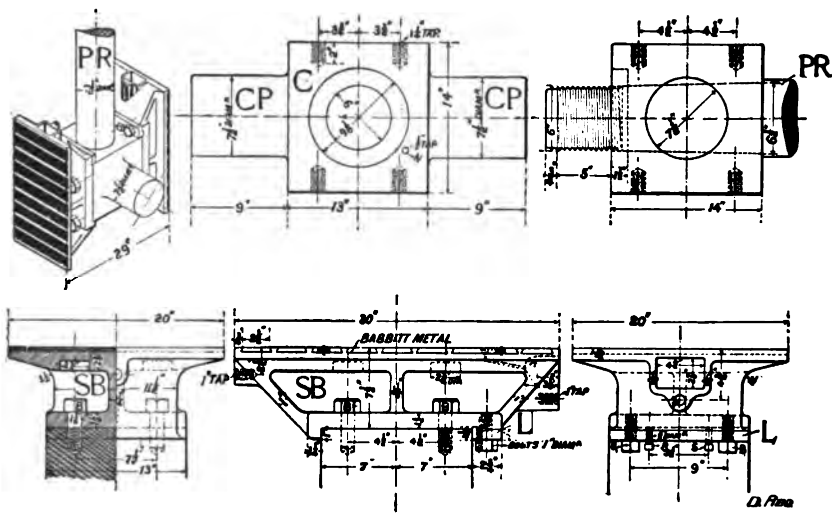


FIG. 298.

Exercise 148.—Draw a general arrangement of the cross-head shown in Fig. 299. Show a FRONT ELEVATION, a HALF PLAN, and a HALF SECTIONAL PLAN of the top, the plane of section passing through the centre of the wrist-pin. *Scale 4 inches to the foot.*

Eccentrics.—The eccentric is a form of crank in which the radius of crank-pin is greater than the sum of the radii of the crank and the shaft, as shown in Fig. 300, where the

crank is shown by dotted lines, and the eccentric by full lines. It is used for converting circular into reciprocating motion. For this purpose its action is identical with that of a crank, and as the eccentric absorbs more power than the crank (owing to the greater leverage at which the friction acts) it is used in preference only where the throw is comparatively

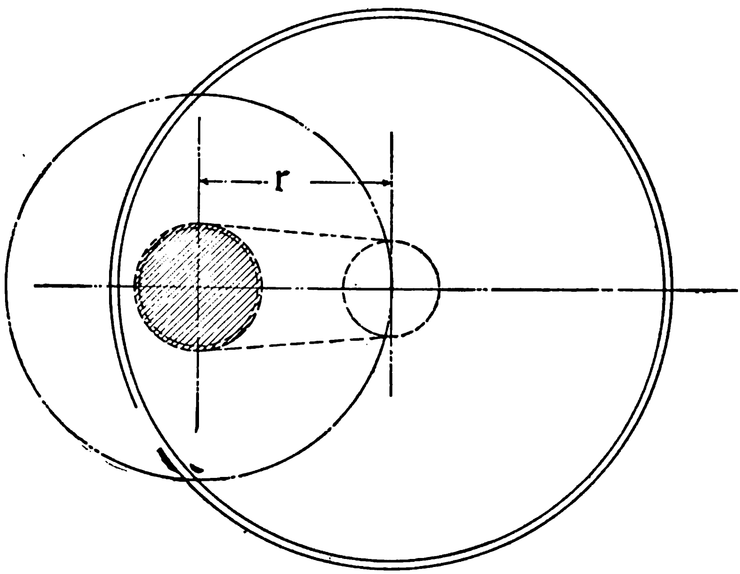


FIG. 300.

short. The eccentricity or throw of the eccentric is the distance *r* from the centre of the shaft to the centre of the sheave. The stroke of the reciprocating piece worked by the eccentric is equal to twice the throw.

Fig. 301 represents an eccentric used for working the slide-valve of a locomotive engine. The eccentric proper is generally called the sheave or pulley. When it cannot be passed on to position over the end of the shaft, the sheave is

made in two parts, P and P', parted on a line passing through the centre of the shaft and at right angles to the horizontal centre line of the eccentric, and held together by studs. That

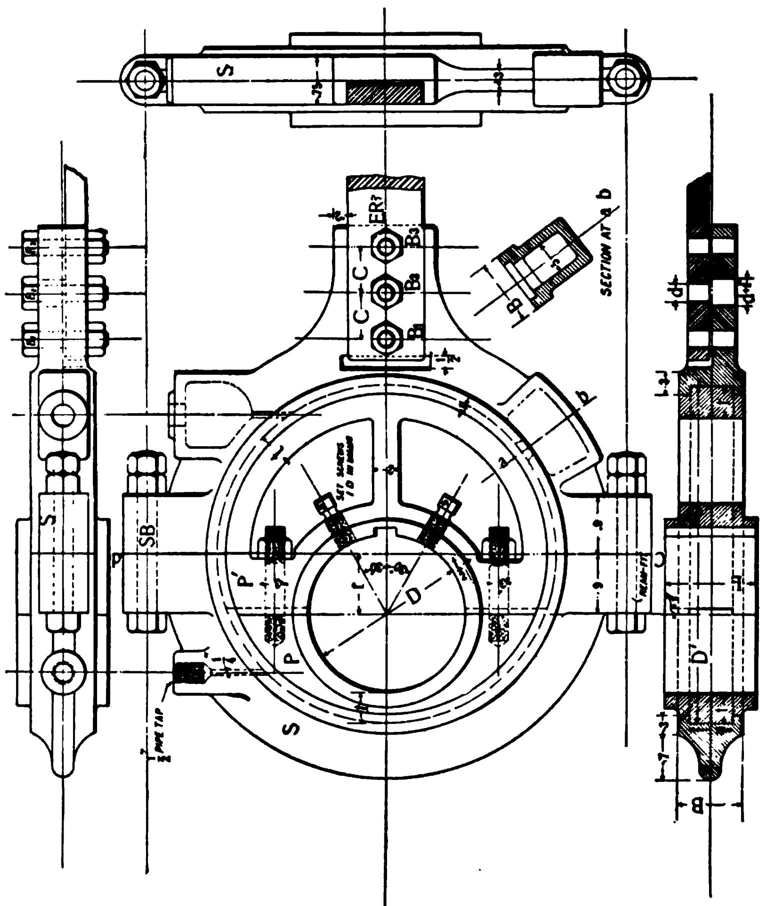


FIG. 301.

the strain may come on the stronger part, P', the key and set-screws used in fastening the sheave to the shaft are placed on that part. The eccentric-rod ER is secured to the strap S bv

the bolts B_1 B_2 B_3. The hole through the strap, for the centre-bolt B_2, is elongated that the rod ER may be adjusted when setting the valve.

Proportions. — The thickness t of the sheave may be $\frac{1}{4}D - \frac{1}{8}''$, with a minimum of $\frac{1}{2}''$. The diameter of the sheave will then $= D + 2r + 2t$. The breadth B of the sheave may be found by the formula

$$B = \frac{L}{D' \times p},$$

where L = load driven by the eccentric;

D' = diameter of the sheave;

p = allowable pressure per square inch of projected

f_t = area, which should have a maximum of 100 lbs.

Thickness of key = $.1D$.

Breadth of key = $1\frac{1}{2}$ times the thickness.

The size of the strap-bolts SB should be proportioned to resist the load driven by the eccentric.

$$d_1 = \sqrt{\frac{L}{\frac{\pi}{4} \times 2 \times f_t}},$$

where d_1 = diameter at the bottom of the threads;

L = load driven by the eccentric;

f_t = safe strength of bolts, which may be taken at 2000 lbs. per square inch.

The size of the rod-bolts, assuming the load is resisted by the two fitted bolts, may be found by the formula

$$d = \sqrt{\frac{L}{\frac{\pi}{4} \times 2 \times f}}.$$

f may be taken at 3000 for wrought iron. The distance C between centres may be made $= 3d$.

The parts marked in decimals are proportional to B, the breadth.

Exercise 149.—Draw the arrangement of eccentric-sheave and strap shown in Fig. 301, proportioned to carry a load of 2300 lbs., taking the pressure per square inch of projected area = 50 lbs.

Draw the views shown in Fig. 301; also a SECTIONAL END VIEW looking towards the right, the plane of section passing through the eccentric at the line *cd*. Make the eccentric-rod *ER* $3\frac{1}{2}'' \times 1''$. *Scale half size.*

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